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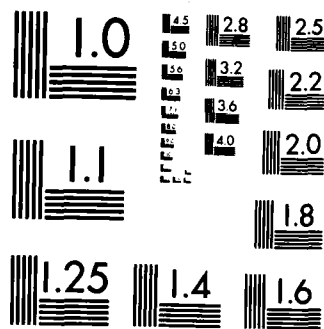
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BENEFITS OF SPIRALLY ENHANCED TUBES FOR STEAM CONDENSERS IN
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DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Maryland 20084



BENEFITS OF SPIRALLY ENHANCED TUBES FOR STEAM
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COOLING SYSTEMS

by

Donald T. Knauss

and

Raymond W. Kornbau

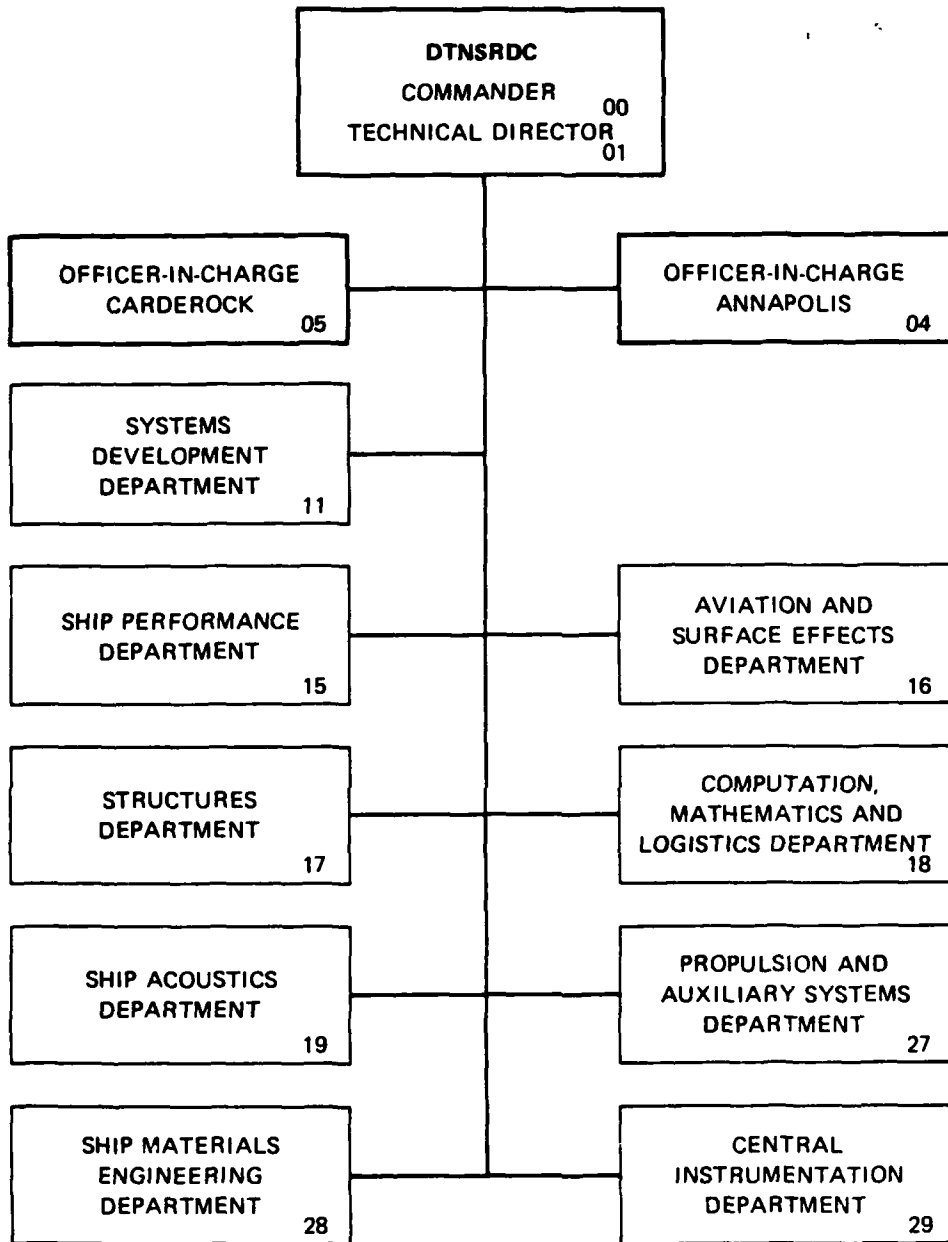
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vacuum. The systems incorporate new high-strength, lightweight materials (titanium and Inconel) and include a main condenser consisting of titanium tubes having a 5/8-in. (1.59 cm) o.d. and 0.035-in. (0.089 cm) wall. General coolant-pumping-power correlations are developed for the spirally enhanced geometries examined. Results for two enhanced-tube geometries indicate that condenser-tube enhancement can provide system weight reductions ranging from 20 to 5 percent for short and long condensers, respectively.

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NOMENCLATURE

A	=	Heat-transfer surface area, ft ² (m ²)
D	=	Condenser diameter, ft (m)
F	=	Tubing enhancement factor; ratio of heat-transfer coefficients (enhanced over smooth), h_e/h_s
G	=	Mass velocity; product of density and velocity of fluid, lb _m /ft-s (kg/cm-s)
H	=	Head loss, ft H ₂ O (kPa)
I	=	Height, ft (m)
L	=	Condenser length, ft (m)
N	=	Bundle tube count
Nu	=	Nusselt number
P	=	Pumping power, hp (kW)
Q	=	Volume flow rate, gal/min (ℓ /s)
Re	=	Reynolds number
S	=	Distance between support plates
$\Delta T \ell_m$	=	Logarithmic mean temperature difference, F (C)
U	=	Overall coefficient of heat transfer = $q/(A \Delta T \ell_m)$, Btu/h ft ² F (W/m ² C)
V_t	=	Flow velocity in tube, ft/s (m/s)
Ψ	=	Machinery volume, ft ³ (m ³)
d	=	Diameter of tube
e	=	Groove depth for a spirally grooved (roped) tube surface
f	=	Friction factor
h	=	Heat-transfer coefficient, where $h = q/(A \Delta T)$, Btu/h-ft ² F (W/m ² C)
i.d.	=	Inside diameter, in. (cm)
ℓ	=	Width of peripheral steam lane, ft (m)
m	=	Mass flow rate, lb/h (kg/h)

n	=	Number of support plates
o.d.	=	Outside diameter, in. (cm)
p	=	Groove pitch; peak-to-peak distance between ridges on a roped-tube surface
q	=	Heat-transfer rate, Btu/h (kW)
r	=	Absolute roughness of tube, ft(m)
t	=	Number of helical starts on roped tube
v	=	Specific volume, ft ³ /lb (m ³ /kg)

GREEK LETTERS

α	=	Aspect ratio, L_{ov}/D_{ov}
δ	=	Number of internal fins
θ	=	Helix angle

SUBSCRIPTS

B	=	Tube bundle
C	=	Condenser
D	=	Darcy
LP	=	Seawater loop, consisting of hull valves, piping, and auxiliary heat exchanger
P+M	=	Pump and motor
S	=	Overall system
W	=	Waterbox
en	=	Enhanced
i	=	Inside
l	=	Liquid
max	=	Maximum
o	=	Outside

ov = Overall condenser
 ()_o = At optimum α
 s = Steam
 sm = Smooth

LIST OF METRIC CONVERSIONS

1 ft = 0.305m
 1 lb/h = 0.454 kg/h
 1 ft³/lb = 0.0624 m³/kg
 1 ft³ = 0.0283 m³
 1 lb = 0.454 kg
 1 psi = 6.9 kPa
 1 in. = 2.54 cm
 1 lb/ft³ = 16 kg/m³
 1 ft/s = 0.305 m/s
 1 gal/min = 0.0631 *L*/s
 1 ft H₂O = 2.99 kPa
 1 hp = 0.746 kW

ABBREVIATIONS

BTU = British Thermal Unit
 CP = Commercially Pure
 NPS = Naval Postgraduate School, Monterey, California
 OU = Operational Unit
 TFCEP = Thin Film Condensate Promoter
 YIA = Yorkshire Imperial Alloys, Leeds, England

ABSTRACT

A design study determined the weight and volume benefits of heat-transfer augmentation in large steam-condenser cooling systems. By using an advanced Navy computer model and the observed dependence of system performance on coolant flow rate, flow-optimized designs for minimum system size or weight could be found for any given enhancement. The analysis, conducted for a given steam flow, assesses the effects of tube geometry, overall condenser length, and design constraints relating to waterbox diameter and condenser vacuum. The systems incorporate new high-strength, lightweight materials (titanium and inconel) and include a main condenser consisting of titanium tubes having a 5/8-in. (1.59 cm) o.d. and 0.035-in. (0.089 cm) wall. General coolant-pumping-power correlations are developed for the spirally enhanced geometries examined. Results for two enhanced-tube geometries indicate that condenser-tube enhancement can provide system weight reductions ranging from 20 to 5 percent for short and long condensers, respectively.

ADMINISTRATIVE INFORMATION

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Special recognition is given to Profs. P. J. Marto and R. H. Nunn (Naval Postgraduate School) for their invaluable technical guidance and support in developing the computer code for complex modeling of the enhanced-tube bundle.

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INTRODUCTION

The technology base relating to enhanced heat-transfer surfaces in general, and condenser enhancement in particular, has been expanding quite rapidly in recent years. The appearance of enhancement techniques other than those of the extended-surface variety has generated considerable interest in the Naval community, where such geometries might offer low-cost options for significantly improving vapor-side and liquid-side heat transfer in both refrigerant and steam systems for shipboard use.

One of the factors contributing to this interest is that the steady evolution of enhanced surfaces, which has been occurring in the refrigeration industry since the 1930's, has finally spawned enhancement techniques which may be applied to steam systems. Other developments have been derived from private industry's increasing emphasis on energy conservation and novel approaches to reducing the construction and operating costs of processing plants. For the first time, domestic and international power stations and distillation plants are starting to employ enhanced-tube surfaces in large steam-condenser bundles.^{1, 2}

During the latter part of the last decade, both the British and U.S. Navies initiated basic research studies aimed at improving shipboard steam-condenser design and have developed an expertise relating to the analysis of large-bundle flow dynamics, which is critical to the accurate prediction of overall condenser performance and size requirements.

The impetus provided by the above developments led to the initiation of a Navy program which would utilize this new information in assessing the benefits of enhanced heat-transfer surfaces in an appropriate model of a shipboard-steam-plant cooling system. Primary attention has been given to minimization of cooling-system weight and volume under a typical set of operating conditions. Weight, volume, and pumping power of the overall system were regarded as the primary performance criteria.

Tube-Bundle Modeling

This task was initiated at the NPS by conducting a survey of nonproprietary state-of-the-art methods for sizing condenser tube bundles. This work resulted in an improved computer code* capable of modeling complex steamside flow dynamics, viz., the effects of condensate inundation, vapor-velocity shear, noncondensable gases, and pressure drop in a two-dimensional steam flow.¹⁵ The fluid dynamic and thermal processes within the tube bundle are modeled two dimensionally, with the steam flowing radially inward toward a central void. The code models the bundle with an equivalent single-pass coolant flow and uses the independent variables relating to tube layout and steam and coolant conditions to calculate the effective bundle length and diameter. A single-pass bundle of semicircular cross section, bounded by the vertical plane of symmetry, is analyzed by dividing it into six equal sectors extending over 30 degrees of arc. For each sector, the calculation procedure is executed on a row by row basis beginning with the outer row and continuing radially inward to the central void, which serves as a collection header for noncondensable gases prior to their passage through the air-cooler section (See Figure 2). The overall performance of the average tube within a sector row is calculated along with the heat flux per foot of tube and the condensation rate. These quantities are then used in generating the inputs to the next tube row.

If, after a pass through the bundle, the specified conditions for exit steam fraction and steam velocity are not satisfied, the bundle length is adjusted and another iteration through the bundle is executed and repeated until the exit conditions are satisfied. In addition to tube layout, the primary program inputs include the steam inlet conditions, exit steam fraction, seawater inlet temperature, and either the seawater flow or seawater velocity

* Elements of this code were originally developed by the Oak Ridge National Laboratory¹⁴ and possessed a unique advantage in their accommodation of enhancement factors to evaluate the overall thermal resistance between the steam and seawater.

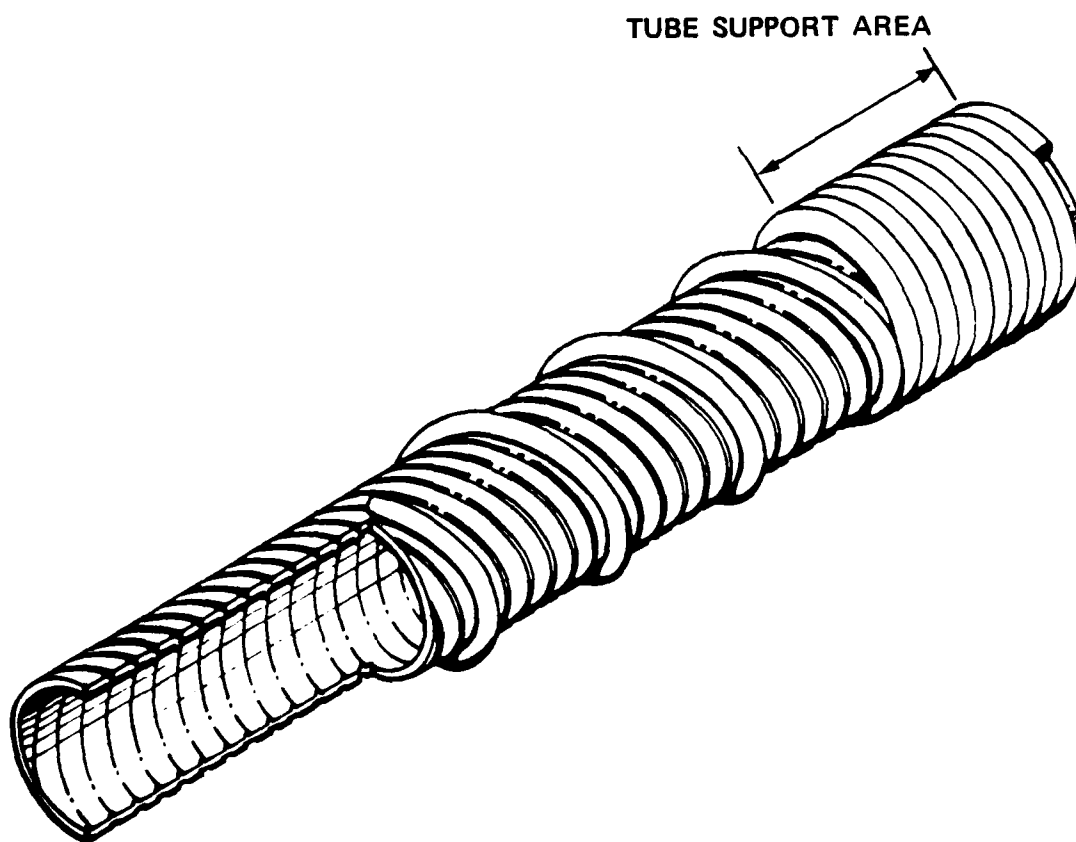


Figure 4 - Illustration of Tube Y with its TFCP in Place
(Denoted as Tube YS)

The Linde Division of Union Carbide Corporation has made a considerable effort to develop the basic concept for smooth-tube bundles in the U. S. power industry.¹³ With this technology base, full-scale, Naval-condenser development costs for the steamside wire wrap would be minimal. Where smooth lands must be provided for roped-tube support, the wire wrap can be collapsed into a continuous coil to provide a load-bearing surface (see Figure 4). As shown in Figure 4, a TFCP-equipped roped tube with a wire-coil pitch close to existing condenser test data¹³ could be achieved by simply wrapping only one of the six helices on tube Y. In addition to enhancing performance, the roped substrate of tube Y also facilitates the TFCP installation and enhances the structural integrity of the wire/tube system. To differentiate between tube Y and its TFCP-modified version, the designation YS has been applied to the latter.

While the internal performance of tube YS can be derived from Equations (1) and (2), the steamside performance is based on both Equation (3) and the observations of Czikk et al.⁹, who showed that the double enhancement of the exterior surface is achieved by elimination of the inundation effects due to condensate drainage within the tube array; i.e., the double enhancement does not augment the single-tube performance but simply serves as a device for preventing the surface inundation that occurs when the condensate leaving one tube impinges on another tube below it. Within a large bundle, this flooding of the tube surface reduces the overall heat-transfer significantly. However, test data clearly indicate that the TFCP is capable of maintaining single-tube performance on tubes that are at least 30 rows below the top tube. Since this configuration approximates the bundles examined in this study, the effect of condensate rain on heat transfer was neglected in modeling tube YS. A detailed discussion of techniques for using the TFCP on roped tubes in an operational main condenser is given in the Appendix.

option; therefore, an evaluation of the tube-sink penalty was included in this study.

TABLE 1 - PERFORMANCE CHARACTERISTICS OF ENHANCED-TUBE Y

t	p/d _i	e _o /d _i	f _D	F _i	F _o
6	0.150	0.0209	0.104 *	2.16	1.41

* mean value over $Re = (4.0 - 6.0) \times 10^4$

The number of helical groove starts can affect not only tube sink, but also heat-transfer performance. This is most likely due to the influence of helix angle on condensate drainage. In view of the very limited amount of experimental data related to start effects and the inconsistent conclusions derived from them^{8, 11, 12} a reliable prediction of these effects for low-pitch profiles is not possible. More test data is required to fully ascertain the penalty associated with condensate drainage when the number of starts increases.

As previously discussed, the tube roping profile was altered to achieve the steamside heat-transfer enhancement afforded by the Linde TFCP. Through consultation with Union Carbide Corporation, the TFCP fabrication procedure was modified to improve the integrity of the promotor in a shipboard environment. In this process, a wire of appropriate size and material is impressed into the helical grooves to extend the steamside concave-convex drainage profile. By the accelerated movement of condensate toward the wired channels, the surface-tension forces promote thinning of the condensate layer between the wires. (See Figure 4).

calculation of both the internal friction and heat transfer from given external characteristics. Quantitatively, the Yorkshire correlations for the selected roped tube yield the following Darcy friction factor and enhanced heat-transfer coefficients:

$$f_D = 0.239 \text{ Re}^{-0.077} , \quad (1)$$

$$h_i = 2.16(h_i)_{sm} , \quad (2)$$

$$\text{and } h_o = 1.41(h_o)_{sm} . \quad (3)$$

The above value for the outside heat-transfer coefficient h_o has been checked against empirical correlations developed by Catchpole and Drew¹¹ for a variety of high-performance roped tubes. These correlations account for the surface tension and condensate-drainage phenomena that control exterior performance. The resulting enhancement factor was in exact agreement with the value in Equation (3). The characteristics of tube Y, whose performance was found to be markedly superior to that of other vendors' products, are presented in Table 1.

It is important to note that the production of profiles similar to tube Y in some high-strength materials, such as titanium, may incur a slight performance penalty due to the occurrence of tube sink; i.e., the outside diameter of the tube shrinks slightly during the roping process. Moreover, the amount of this sink is dependent upon the number of helical groove starts. For a titanium tube Y of 5/8-in. (1.59 cm) o.d. X 0.035-in. (0.089 cm) wall, the o.d. sink is assumed to be about 14 mils (0.36 mm) for 3-start roping and 11 mils (0.28 mm) for 6-start roping; for a given tube-sheet solidity and thickness, such sinkage could conceivably have a marked effect on large-bundle, heat-transfer area. Since tube spacing has a major impact on tube-sheet and condenser strength, alteration of the tube-sheet solidity was not considered a viable

applied to the enhanced tube. In the smooth-tube evaluation, the finite-element stress analysis was applied to the short bundle end sections bounded by the inner and outer tube sheets since the most severe tube stresses occur in this region. The stress model divided the tube bundle into two outer rows of rigid tubes and a large inner tube volume of soft material; i.e., the inner tubes formed an elastic foundation for the more highly stressed outer rows. The forces and moments contributing to the axial stress on the tubes were due to (1) waterbox pressure, (2) bolt preload, (3) thermal deformation, and (4) hydrostatic pressure in the tube. From an analysis of various stress modes, it was concluded that the purely tensile/ compressive stresses were markedly more limiting than either the bending or fatigue stresses, and that the direct stresses could be held within the existing design margin by using a wall thickness of 0.035 in. (0.089 cm).

For a tube having a 5/8-in. (1.59 cm) o.d. and 0.035-in. (0.089 cm) wall, it was then possible to utilize the performance correlations of one of the leading manufacturers of roped titanium tubing to develop a candidate geometry which offered the best tradeoff of overall heat transfer, internal frictional resistance, and ease of fabrication. The resulting roped tube geometry is designated "Tube Y," after its manufacturer—Yorkshire Imperial Alloys (YIA) of Leeds, England. The Yorkshire procedure⁸ yielded a profile which would maintain an interior enhancement comparable to that offered by other tube vendors, while achieving an exterior enhancement far superior to all other known profiles. This, however, could only be achieved by accepting a somewhat higher friction penalty. The YIA correlations, which permit direct calculation of enhanced-tube friction as a function of Reynolds number, differentiate between inside and outside groove depth. The ratio of these depths, given as a function of the tube wall thickness/pitch ratio, is essential to the

did not yield substantial performance gains when the biofouling risk was considered.

Additional roped-profile modifications may also be obtained by superimposing special surface-tension promoters onto the steamside surface to further enhance the heat transfer. However, very few highly effective techniques for such double enhancement of the steamside appear to exist; therefore, the lack of commercial experience has limited consideration to either a helical wire wrap [Thin-Film Condensate Promoter (TFCP)]⁷ or a series of helical microgrooves which are fully integrated into the basic rope profile. Since the latter technique has not yet been successfully applied to titanium, the TFCP remains as the only potentially cost-effective method for amplifying the external performance of a roped tube.

Tube Design and Performance

The friction factors and heat-transfer coefficients (interior and exterior) for the above tubes were obtained from vendor empirical correlations^{8, 9} and experimental data developed independently at the Naval Postgraduate School (NPS)¹⁰. The heat transfer of the basic rope profile was expressed in the form of an enhancement factor which could be applied to the calculated heat-transfer coefficients for the equivalent smooth tube operating at the same flow conditions. A review of the existing empirical data indicated that the enhancement factors were independent of tube diameter and Reynolds number and were only a function of p/d and e/d , where p is the groove pitch, e is either the internal or external groove depth, and d is either the internal or external tube diameter.

In order to evaluate the above performance parameters, a nominal marine-condenser tube of 5/8-in. (1.59 cm) o.d. was selected. It was concluded that the wall-thickness estimate found for a similar smooth-tube bundle could be

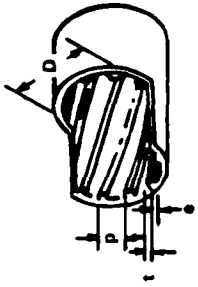
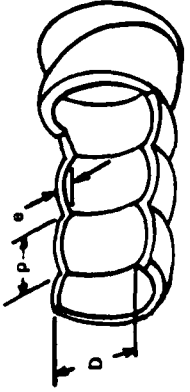
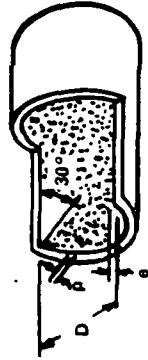
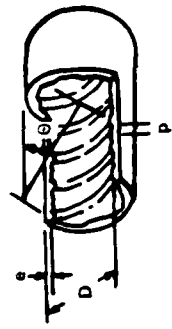
in Figure 3 could greatly affect the rate and extent of fouling and, consequently, the heat-conduction characteristics of the tube.

In this study, the tube material was restricted to unalloyed titanium (commercially pure, grade 2); consequently, many excellent heat-transfer geometries that were developed for copper tubing in the refrigeration industry were rejected because of present tube-manufacturing limitations of titanium.* Even if the more exotic copper-tube profiles were produced in titanium, biofouling in the seawater environment appears more likely in the larger crevices where flow stagnation may occur.

One profile that has been successfully applied to titanium steam-condenser tubes has helically scribed grooves, integrally registering on both the steam and seawater surfaces. This profile simultaneously promotes higher heat-transfer rates on both surfaces by providing waterside turbulence and convex-concave steamside surfaces to drain and carry away the condensate. Because its appearance resembles a rope, this tube type is aptly referred to as a "roped" or spirally grooved tube.

This profile was selected over other candidates in Figure 3 as the lowest-risk option for the demanding structural and environmental requirements of a Naval combatant. An important advantage of the roped tube is that its profile can be varied to maximize heat transfer for a wide range of seawater and steam conditions; the ability to provide geometries with a wide variation in the number of continuous helices applied to the surface (number of helical starts), as well as wide variations in pitch and groove depth, has been firmly established.^{5, 6} The development costs associated with other enhancement geometries

* Extensive cold working of CP-2 titanium produces high stress concentrations from work-hardening, which, in turn, would require annealing. This approach has not been developed for cost considerations.

TYPE ROUGHNESS	GEOMETRY	RANKING	CHARACTERISTIC DIMENSION *
FIN		1	δ, θ $e/D, e/t$
ROPED		2	$e/D, p/D$
3-D		3	$e/D, p/D$
RIB		4	$e/D, p/D, \theta$

* t = fin thickness

Figure 3 - Candidate Tube Enhancements; Waterside Performance

higher heat-transfer rates have been achieved through greater turbulence and pressure drop of the cooling liquid at a given velocity. Recent studies⁴ have identified tube geometries which provide the maximum gains in heat transfer over smooth-i.d. surfaces for shell-and-tube surface condensers (see Figure 3). The ultimate selection from among these candidates, however, must also depend on other considerations relating to the overall optimization of both the steamside and waterside profiles of the tube, and the manufacturability of a tube having such profiles. In general applications, the effectiveness of the geometries of Figure 3 reduces the dominance of the waterside resistance and directs attention to the additional benefits of a corresponding steamside improvement.

The tube's steamside condensing heat transfer can be enhanced by either extended surfaces, dropwise condensation (achieved by a thin non-wetting surface coating) or by altering surface-tension forces in the condensing liquid film. The latter approach involves surface alterations which can modify the forces in two different ways; on convex surfaces, the liquid-film thickness is reduced, which reduces resistance to heat transfer, or, alternatively, on concave surfaces, the film thickness is increased, thus enabling the tube to shed liquid more quickly. This type of geometry is illustrated in Figure 3 as a roped tube.

Ideally, maximum heat transfer can be achieved when both the internal and external surface geometries can be independently varied to meet the individual requirements of the water and steamside conditions. Unfortunately, this is often costly from a tube manufacturing standpoint, especially for titanium or other high-strength alloys in which large residual stresses are imposed from cold working. Another major consideration with titanium tubes is seawater bio-fouling. The flow turbulence generated by the various enhancement geometries

(2) Auxiliary Heat Exchanger. This unit utilizes smooth titanium tubes and titanium and Inconel 625 for the shell and waterboxes, respectively.

(3) The MSW-pump casing consists of Inconel 625.

(4) The Hull-valve body material is assumed to be HY-80 for flows under 8000 gal/min (505 ℓ /s) and HY-130 for flows above 8000 gal/min.

(5) Piping consists of Inconel 625 throughout.

The above specifications apply to both the smooth-tube (baseline) and enhanced-tube cooling systems developed in this study.

CONDENSER DESIGN ANALYSIS

For the nominal condenser heat load selected for this study (2.03×10^8 Btu/h or 59,500 kW), a 26-ft (7.93-m)-long condenser operating at a pressure of 6.0 in. Hg (20.3 kPa) would require about 3000 smooth tubes to condense the exhaust steam leaving the plant's turbines. The tube surface area required for this heat duty is inversely proportional to the conductance across the tube wall. For smooth tubes this can amount to over 10,000 ft^2 (930 m^2) of heat-transfer area. Improving the efficiency of heat transfer by modifying the geometry of the tube surface can substantially reduce this area and the consequent size and weight of the tubes, shell, waterboxes, and numerous supporting foundations.

Tube Selection

In shell-and-tube, steam condenser design, most development has been directed toward increasing the heat-transfer coefficient between the tube-side cooling water and the tube wall. A great variety of geometries has been studied³, including tube-wall spiral ridging (roping), longitudinal ribs or fins, helical ribs or fins, repeated ribs, etc. In all of these geometries,

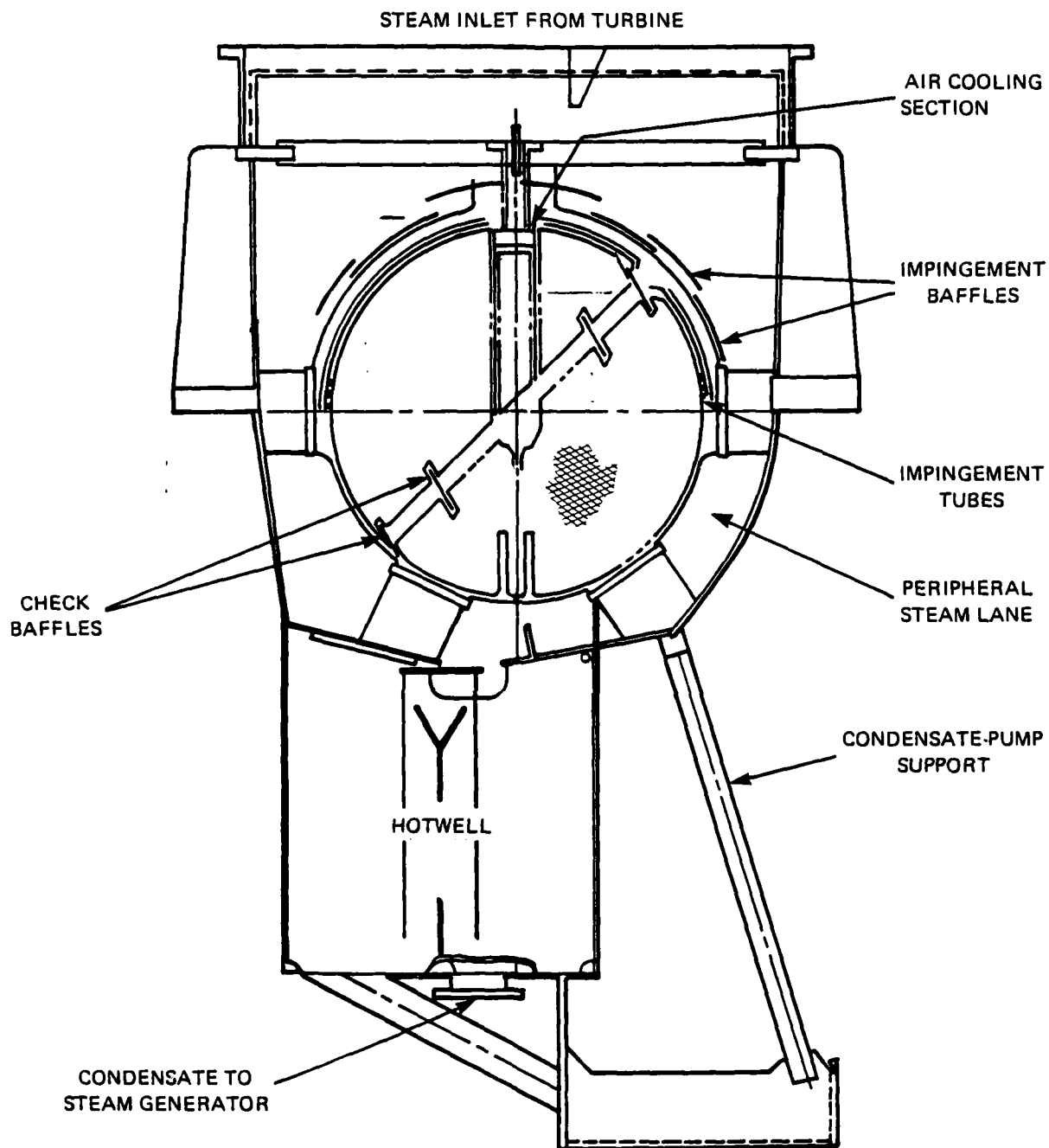


Figure 2 - Cross Section of the Operational Main Steam Condenser Selected for Material and Tube - Geometry Modification

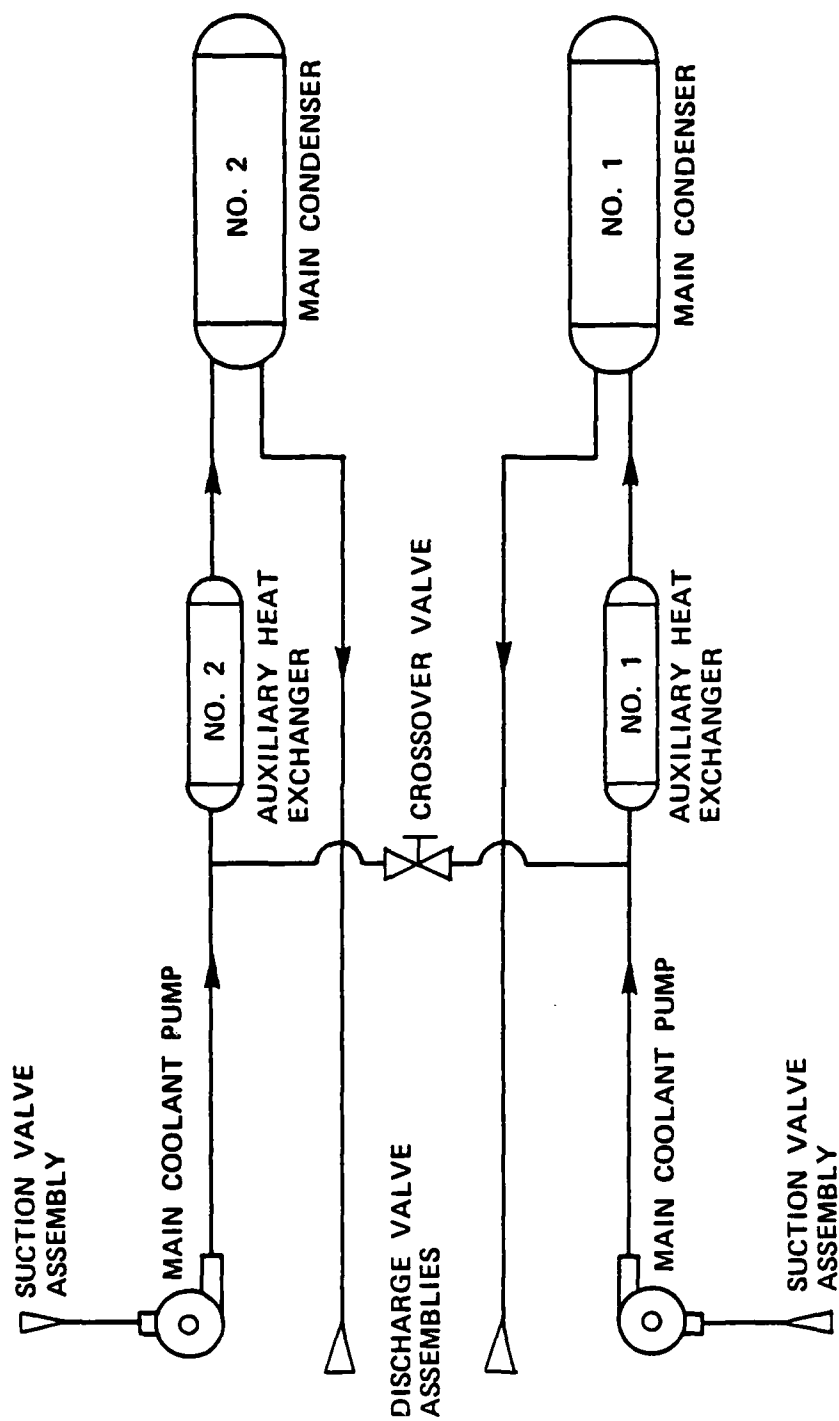


Figure 1 - Flow Schematic of a Typical Main Cooling System for a Steam Propulsion Plant

APPROACH

COOLING-SYSTEM DESCRIPTION

The main cooling system employed in Naval-propulsion-machinery compartments accounts for approximately half of all the machinery located in those spaces. The function of this system is to reject heat from both the main steam condenser and an auxiliary heat exchanger, the latter serving to remove heat from fresh-water-cooled equipment within the compartment. The main condenser, which is one of the largest single machinery components aboard ship, constitutes, with its foundations, nearly 2/3 of the total main-cooling-system weight or volume. Any modification of the condenser operating conditions has a major impact on the entire cooling-system size and weight since the amount of coolant needed to reject the heat will determine the size of the piping, pumps, valves, etc. The interrelationship of the major components impacting the system size is shown schematically in Figure 1.

In addition to specifying the operating conditions of the cooling system, it is also necessary to establish the kinds of materials to be used in its fabrication. The materials selection was strongly influenced by the desire to reduce system weight; therefore, materials with high strength-to-weight ratio (titanium and inconel) were selected over more traditional copper-nickel marine alloys. A general description of the major cooling-system components is included here to reflect these design improvements.

(1) Main Condenser. The operational main condenser which served as a reference in developing the baseline unit is shown in Figure 2. In its upgraded form, the condenser consists of smooth titanium tubes. Peripheral steam-impingement baffles and tubes have been eliminated. The steel shell and inner tube sheets are flanged with titanium outer tube sheets and Inconel-625 waterboxes.

in the tubes. All of these parameters will affect bundle size, and, except for tube count and seawater velocity, were generally considered to be invariant. Consequently, if a given bundle length or diameter is desired, it can only be found by a trial and error process which adjusts either the tube count or seawater velocity.

A significant feature of the NPS computer code, which is identified by the acronym MORCON, is a new steamside heat-transfer correlation which accounts for what is termed vapor-shear effects; i.e., the shearing of the tube's condensate layer by the high-velocity vapor--a condition prevalent in marine condensers where high condensation rates occur. Since MORCON determines the "average-tube" performance for each sector row, the formulation due to Fujii et al.¹⁶, which is based on the flow past a single tube, is quite appropriate. The uniform-heat-flux correlation, when normalized with respect to the Nusselt prediction (Nu_o) for quiescent vapor undergoing free convection over a tube at uniform temperature, may be written

$$\frac{Nu}{Nu_o} = 1.24 Nu_o^{-0.2} Re_1^{0.1} \quad (6)$$

where

$$Nu_o = 0.725 \left(\frac{g \rho_1^2 d_o^3 h_{fg}}{k_1 \mu_1 \Delta T} \right)^{1/4}$$

$$Re_1 = \frac{\rho_1 U_\infty d_o}{\mu_1}$$

and

- U_∞ = free-stream vapor velocity
- d_o = outside diameter of tube
- h_{fg} = latent heat of vaporization
- ρ_1 = density of condensate layer
- g = local acceleration of gravity
- k_1 = thermal conductivity of condensate layer
- μ_1 = dynamic viscosity of condensate layer
- ΔT = temperature difference across the condensate layer

The appropriate value of U_{∞} to apply to a tube within the sector is somewhat controversial. The velocity actually seen by such a tube is probably most closely approximated by the velocity based on the minimum flow area between adjacent tubes in a row. The conditions at this throat area are easily calculated and used in determining the pressure drop due to shear.

One other important provision in the MORCON code is a correction for condensate inundation. This correction, which is applied to the average tube in each sector row, is a function of the number of tubes directly above the tube being evaluated. The condensate shedding from each tube impinges on the tube directly below it and contributes to flooding of the lower tube rows. This flooding effect, which thickens the condensate layer, causes the thermal resistance of the tube to increase above the value for a single isolated tube. In developing a correction for condensate rain, it is important to note that the rain is affected by vapor velocity and direction and by tube-bundle layout. The most widely accepted empirical model for predicting inundation effects is able to account for side-drainage---a phenomenon in which surface tension effects cause some of the condensate to leave a tube sideward and flow unto adjacent tubes rather than straight down to the tube below. A more detailed discussion of the MORCON code and tube-bundle effects is given by Nunn and Marto¹⁵.

The selection of an appropriate tube spacing must also be made in the bundle design process. This is an extremely critical design parameter since it controls the solidity of the tube sheets, which anchor the tubes at either end of the condenser and provide an interface with the waterboxes. Since the void fraction (defined as hole area divided by total plate area) determines the structural integrity of the tube sheet, design specifications usually state a definite upper limit for this parameter¹⁷. In the present study, the applied value corresponds to that used in an operational smooth-tube design (Figure 2);

this value is assumed to represent the design limit.

Condenser Shell Modeling

The material and design characteristics of the main-condenser shell were combined with the MORCON tube-bundle size predictions to determine overall condenser weight and volume. In sizing the circumferential steam lane on the outer periphery of the bundle (Figure 2), an average steam-lane velocity of 160 ft/s (49 m/s)* was assumed. In sizing a condenser for a given bundle size or steam condition, the lane width was adjusted to maintain this velocity. For this formulation, the lane width (ℓ), in feet **, can be written

$$\ell = 5.17 \times 10^{-7} \left[\frac{\dot{m}_s v_s}{(L_B - 1.84)} \right] \quad (7)$$

where

\dot{m}_s = mass flow of steam, lb/h

v_s = specific volume of steam, ft³/lb

and L_B = effective length of the bundle (between tube sheets), ft

The modeling of the overall condenser is achieved by expressing the various component weights and volumes in terms of the condenser operating conditions. The identification of the major components for inclusion in this condenser buildup is facilitated by the exploded view of Figure 5. The box volume of the condenser is assumed to consist of the box volume of the circular shell plus the box volume of the hotwell. The hotwell volume is assumed to be proportional to the steam mass-flow rate, and the operational condenser is taken as a reference for estimating hotwell volume.

* Based on the total flow area on a diametral plane of symmetry through the bundle

** Consult list of metric conversions at beginning of text

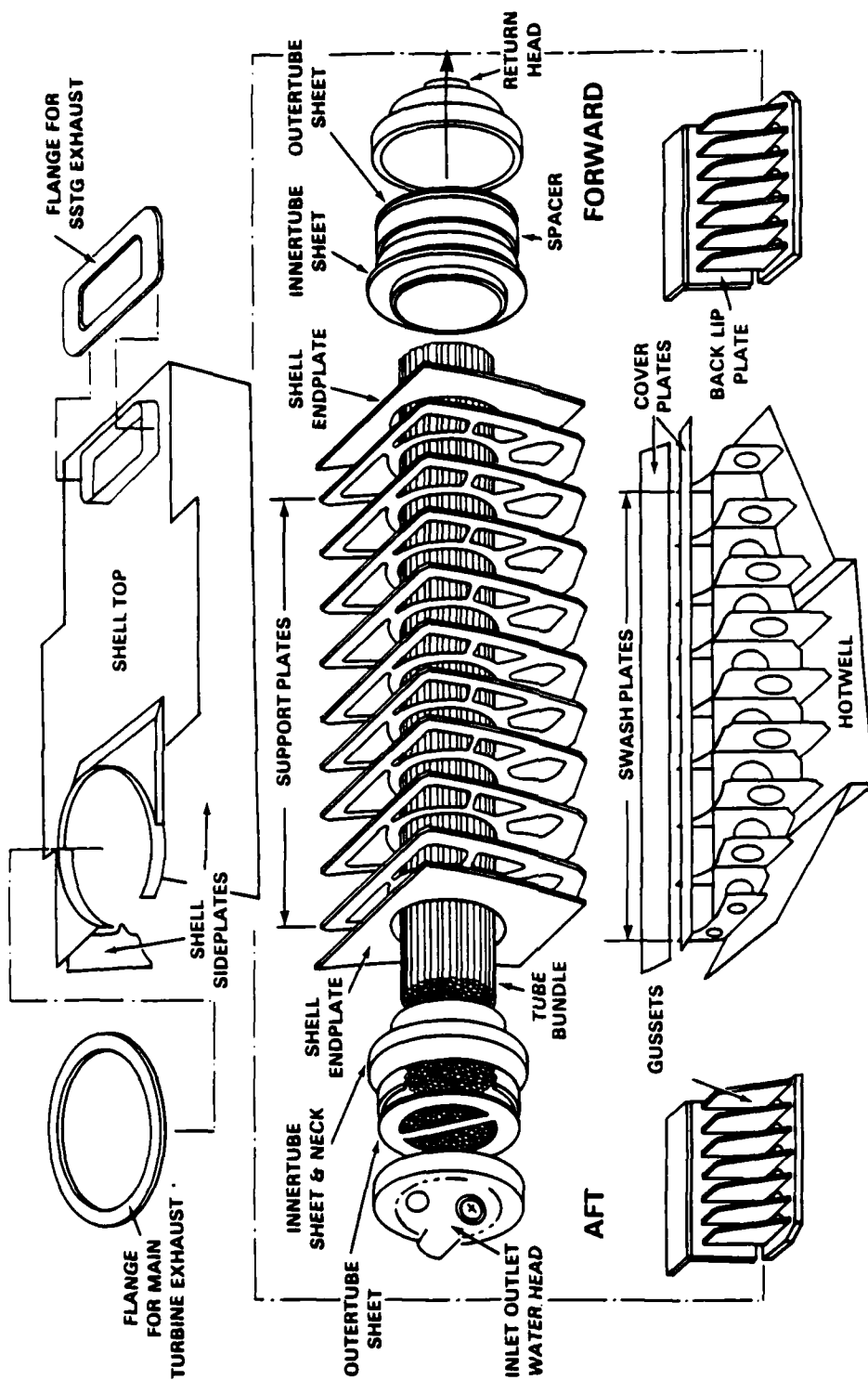


Figure 5 - Exploded View of Main-Condenser Shell Structure

After expressing the above volumes in terms of dimensions, it can be shown that the condenser box volume (with hotwell), in cubic feet, is given by

$$V_C = D_{ov}^2 L_{ov} + 9.2 \times 10^{-4} \dot{m}_s \quad (8)$$

where

$$L_{ov} = \text{overall length} = L_B + L_W + 1.9, \quad (9)$$

$$D_{ov} = \text{overall width} = D_B + 0.3 + 2\ell, \quad (10)$$

with D_B denoting bundle diameter, in feet. The contribution of the waterboxes to the condenser length is

$$L_W = D_B + 0.3; D_B < 7.5 \text{ ft (2.29m)} \quad (11a)$$

$$= 7.5; D_B \geq 7.5 \text{ ft} \quad (11b)$$

Moreover, the overall height of the condenser (with hotwell), in feet, is given by

$$I_{ov} = D_{ov} + \frac{0.0019 \dot{m}_s}{D_{ov}(L_B - 1.84)} \quad (12)$$

The two conditions imposed on L_W are consistent with design limitations on waterbox depth.¹⁷

The condenser weight was found by itemizing all the primary contributions, i.e., the weights that could be explicitly calculated by formula, viz., tube bundle (W_B), condenser shell (W_{SH}), hotwell (W_{HW}), tube support plates (W_{SP}), tube sheets (W_{TS}), and waterboxes (W_{WB}). The sum of these items was then multiplied by a correction factor to account for miscellaneous mounting hardware, such as brackets, bolts, etc. By following this procedure, it can be shown

that, with the addition of fluid weight (W_F), the wet weight, in pounds, of a modified main condenser is

$$W_C = 1.22 \left[\left(\frac{L_B + 1.75}{L_B} \right) W_B + W_{SH} + W_{HW} + W_{SP} + W_{TS} + W_{WB} \right] + W_F \quad (13)$$

where W_B is provided by MORCON,

$$W_{SH} = 102 D_{OV} (L_B - 1.84) \quad (14)$$

$$W_{HW} = 0.45 \dot{m}_s \left[\frac{0.1 n + 0.15}{L_B - 1.84} + \frac{0.19}{D_{OV}} \right] \quad (15)$$

$$W_{SP} + W_{TS} + W_{WB} = (41 n + 1090) (D_B + 0.3)^2, \quad (16)$$

and

$$W_F = 50 N d_i^2 (L_B + 1.75) + 38 (D_B + 0.3)^3 + 0.0574 \dot{m}_s, \quad (17)$$

where N is the number of tubes in the bundle and d_i is the inside diameter of the tube, in feet. The total weight of the primary components ranged from 82 to 87 percent of the total condenser weight; the precise value was determined by the construction materials incorporated into the model (modified or unmodified system).

The number of support plates, n , is obtained by dividing the tube length, L_B , by the required plate spacing. This spacing is obtained from a consideration of tube-bundle hydrodynamic and shock stability requirements. The hydrodynamic stability of both smooth and enhanced tubes was evaluated by the smooth-tube procedure described by Peake et al.¹⁸ The applicability of this method to enhanced tubes is strongly supported by recent studies in this area.^{19, 20}

An additional correction was then applied to provide shock protection equivalent to that of the operational unit. The plate spacing, in feet, found in this approach was

$$S = 0.271 \left[\frac{EIv_s}{V_s^2 d_o} \right]^{1/2} + 0.170 \quad (18)$$

where E is the modulus of elasticity in psi, I is the moment of inertia of the tube cross section in in⁴, v_s is the specific volume of steam in ft³/lb, V_s is the steam velocity in ft/s, and d_o the tube outside diameter in feet.

All of the above equations for the condenser sizing were coded for solution on a programmable calculator.

The accuracy of the above model was assessed by predicting the characteristics of the operational unit (O.U.) in Figure 2. Simulation of this unit required special consideration of the voids which are distributed throughout the shell volume, i.e., several intrabundle steam lanes (including the central and interbank) as well as a peripheral lane. The intrabundle lanes can only be accounted for by applying a correction factor to the bundle diameter calculated by MORCON, which simply adjusts a central void volume until steam exit conditions are satisfied and all tube rows are filled in providing the required tube count. By actual calculation of all inner void areas of the O.U., it is possible to obtain the additional void requirement and the corresponding correction to the bundle diameter calculated by MORCON; this correction increased the bundle diameter by about four percent. The accuracy of the adjusted computer results was determined by comparing them with actual O.U. data. The differences, given in Table 2, show excellent agreement between values. For the evaluation of advanced systems, it was desirable to use a void correction factor which provides exact agreement with the actual O.U. bundle diameter.

A new factor of seven percent was, therefore, applied to all advanced-condenser calculations.

TABLE 2 - ACCURACY OF NPS/DTNSRDC COMPUTER-MODEL PREDICTIONS
FOR AN OPERATIONAL CONDENSER

PARAMETER	PERCENT DIFF.
WEIGHT, lb (kg)	-3.1
VOLUME, ft ³ (m ³)	-4.0
WIDTH, ft (m)	-0.9
HEIGHT, ft (m)	+1.0
LENGTH, ft (m)	-2.5
TUBE SW VELOCITY, ft/s (m/s)	-
NO. OF TUBES	-

COOLING-SYSTEM DESIGN ANALYSIS

Modeling of Loop Components

The overall main-cooling-system weight, volume and head loss were based on manufacturer's data for the hull valves, piping, and auxiliary heat exchangers. At discrete flows of 7900, 10,000, and 12,000 gal/min (498, 631, 757 ℓ /s), mathematical expressions were developed to characterize the weight, volume, and head loss of these loop components as a group; these parameters, when expressed as a function of flow rate, Q , in gal/min, become

$$W_{LP} = 0.053 Q^{1.64} \quad (19)$$

$$V_{LP} = 8.63 \times 10^{-7} Q^{2.31} \quad (20)$$

$$H_{LP} = 70.4 - 2.5 \times 10^{-3} Q, \quad (21)$$

where W_{LP} , V_{LP} , H_{LP} are in units of lb, ft^3 and ft of water, respectively, for the entire ship (two cooling loops). Approximately 23 percent of the auxiliary-heat-exchanger weight was found to be in the foundation. Seawater velocities were maintained within specified acoustical limits of 19 and 25 ft/s (5.8 and 7.6 m/s) for the piping and valves, respectively.

The weight and volume of the pump/motor combination were found by first evaluating the pumping power required to overcome the system head loss (the sum of the condenser and loop losses). The condenser head loss consisted of the frictional loss in the tubes, the tube entrance and exit losses, and the waterbox losses. The tube frictional loss was evaluated by substituting a representative friction factor into the Darcy-Weisbach formula.²¹ In the

Reynolds-number range of interest, the smooth-tube friction factor is most accurately represented by the Colebrook equation²¹, which may be written

$$\frac{1}{\sqrt{f_D}} = -2 \log \left[\frac{r/d_i}{3.7} + \frac{2.51}{Re \sqrt{f_D}} \right] \quad (22)$$

Moreover, for any tube bundle, the head loss, in ft of water, may be expressed

$$H_B = 0.0318 f_D \left(\frac{L_B + 1.75}{d_i} \right) v_t^2 \quad (23)$$

where f_D for enhanced tubes is given by Equation (1).

In the above equations, r is the absolute roughness of the tube (taken to be 5×10^{-6} ft (1.53×10^{-6} m) for drawn tubing), v_t the seawater velocity (ft/s), and Re the tube Reynolds number. Whereas overall evaluation of the remaining condenser losses follows a procedure given by Johnson²², the Crane method²³ was applied to obtain tube entrance/exit losses, and the recommendations of the Heat Exchange Institute²⁴ and Harrington²⁵ were used in determining the in-out-waterbox loss. The return-waterbox loss was then estimated by comparing the predicted and published values for the combined entrance/exit and waterbox losses of the operational condenser. All of the above condenser losses were combined to express the overall loss, in ft of water, as

$$H_C = H_B + 0.031 v_t^2 (\gamma^2 + 0.5 \gamma + 0.66) \quad (24)$$

where

$$\gamma = 1 - N^2 \left(\frac{d_i}{D_B} \right)^4 \quad (25)$$

After determining the total head loss for the cooling system, it was possible to calculate the total pumping-power requirement. If a typical coolant-pump efficiency of 72 percent is applied, the pumping power, in horsepower, for a given flow and system head loss ($H_S = H_C + H_{LP}$) is simply

$$P_S = 3.52 \times 10^{-4} Q H_S \quad (26)$$

In view of the dependence of the coolant-pump speed on flow rate, which, in this study, may be expected to have a very broad range (7500 - 15000 gal/min or 473 - 947 ℓ /s), it became apparent that weight and volume predictions for two different pump speeds would be required.²⁶ The manufacturer's data for pump and motor weight and volume, based on a speed of 1200 rpm, appeared to be valid for flows below about 13,000 gal/min (820 ℓ /s). However, the available pump design data applicable to higher flows were based on a double-suction-volute design and dictated the use of a 900-rpm pump speed²⁶; the impact of this speed change on pump size was found to be insignificant. Although the viability of a double-suction pump may be debatable, a cursory analysis of appropriate single-suction designs for the higher heads that may occur in some enhanced systems (150-200 ft H_2O or 449-598 kPa) suggests that the weights and volumes of these two pump options may not differ significantly. The effect of shaft speed on motor weight was obtained by estimating the weight of a 900-rpm motor at a specified power²⁷ and making the assumption that the variation of motor weight with power at 900 rpm would be similar to the manufacturer's weight/power curve for a 1200-rpm motor. From these data, it was then possible to correlate the weight of the pump/motor assembly in terms of system flow rate, head loss, and pump power. After allocating 20 percent of the

assembly weight to the foundations, these relations for the weight, in pounds, become

$$W_P + M = 1.62 QH_S^{0.07} + 20.7 P_S + 5620; Q < 13000 \text{ gal/min} \quad (27)$$

$$(Q < 820 \text{ } \ell / \text{s})$$

and

$$W_P + M = 2.02 QH_S^{0.07} + 26 P_S + 8960; 13000 \leq Q \leq 15000 \text{ gal/min} \quad (28)$$

$$(820 \leq Q \leq 947 \text{ } \ell / \text{s})$$

From the aforementioned sources of sizing data, it was found that the box density for the pump/motor assembly did not show much dependence on flow rate and was assigned a constant value of 80 lb/ft³ (1280 kg/m³). This value was applied to the calculated weight (excluding foundations) to obtain the assembly volume.

After correcting the condenser weight and volume to account for the twin-condenser support structure, the expressions for cooling-system weight and volume (per ship) may be written

$$W_S = 2.5 W_C + W_{LP} + W_P + M \quad (29)$$

and

$$V_S = 2V_C + (6.56 + 1.47 D_{OV}) L_{OV} + V_{LP} + V_P + M, \quad (30)$$

with units of pounds and cubic feet, respectively.

All cooling-system calculations were coded for solution on a programmable calculator.

Selection of Design Cases

The main-cooling-system operating conditions are set by the main-steam condenser operating conditions. The approach was to determine the cooling-system characteristics of both smooth-tube and enhanced-tube condensers of equal length, with specific length increments taken to be 20, 25 and 30 ft (6.10, 7.62, and 9.15 m, respectively). Table 3 shows the operating conditions applied at two different condenser pressures. Only the 30-ft (9.15 m) length was applied to the Type-2 design. While Type-1 was taken to represent state of the art, Type-2 was included to demonstrate enhancement potential in the higher-power systems that are possible with higher condenser vacuum. The enhancement geometries described above were applied to 5/8-in. (1.59 cm) o.d. titanium tubes with a wall thickness of 0.035 in. (0.089 cm).

TABLE 3 - MAIN-CONDENSER DESIGN CONDITIONS

CONDENSER TYPE	PRESSURE, in.Hg (kPa)	STEAM RATE, lb/h (kg/h)	HEAT LOAD Btu/h (kW)	INLET STEAM QUALITY	SW INLET TEMP F(C)
1	6.0	200,000	2.03×10^8	1.0	66
	(20.3)	(90,800)	(59480)		(19)
2	2.5	200,000	2.07×10^8	1.0	66
	(8.45)	(90,800)	(60650)		(19)

RESULTS AND DISCUSSION

It is of interest to first compare the smooth-tube and enhanced-tube bundle heat transfer for the two conditions of equal pressure drop and equal flow velocity. These comparisons for 25-ft (7.62 m), Type-1 condensers are made in Figure 6, where the reference condenser has smooth tubes carrying a flow at 2 ft/s (3.66 m/s). It is seen that the enhanced-heat-transfer benefit at equal velocity is significantly greater than at equal pressure drop; moreover, the improvement offered by tube YS over tube Y is also better at the equal-velocity condition.

SYSTEM FLOW OPTIMIZATION

It will be shown that when system optimization techniques are applied, neither of the above extreme conditions exists. To minimize cooling-system weight and volume for various coolant flows (flow-optimization method), tube flow velocity and tube count are selected as the independent and dependent variables, respectively, in sizing condensers of fixed length. With this method, it is important to first characterize the relationship between condenser weight and system weight. In Figure 7, this relationship is used to establish the design point for a 20-ft (6.10-m), Type-1, smooth-tube condenser. It is seen that the system weight passes through a well defined minimum at 9710 gal/min (613 ℓ /s), at which point the condenser weight is still falling rapidly. In fact, the condenser weight continues to drop until the coolant flow is well above 11000 gal/min (694 ℓ /s), which corresponds to a tube flow velocity exceeding 10 ft/s (3.05 m/s). These system characteristics permit two possible design approaches—one which minimizes condenser weight or one which minimizes system weight. Comparative studies of the two approaches have shown that a system which incorporates a minimum-weight condenser is four percent heavier

TABLE 6 - CHARACTERISTICS OF OPTIMIZED COOLING SYSTEMS WITH
TYPE-1, TUBE-YS CONDENSERS OF EQUAL HEAT LOAD
(TITANIUM TUBES WITH 5/8" OD X 0.035" WALL)

LENGTH, ft(m)	20 (6.10)	25 (7.63)		30 (9.15)	
DESIGN * CONDITION	W or M	W	M	W	M
TUBE COUNT	2580	1990	2320	1680	2400
TUBE SW VEL., ft/s (m/s)	8.5 (2.59)	10.0 (3.05)	8.0 (2.44)	11.0 (3.36)	7.0 (2.14)
SW FLOW gal/min (ℓ/s)	7940 (501)	7170 (452)	6710 (423)	6710 (423)	6060 (382)
ASPECT RATIO,	3.10	4.70	4.42	6.41	5.65
HEIGHT X WIDTH ft(m)	11.4 X 6.45 (3.48 X 1.97)	9.43 X 5.32 (2.88X 1.62)	9.59 X 5.65 (2.92X 1.72)	8.26 X 4.68 (2.52X 1.43)	8.55 X 5.31 (2.61X 1.62)
VOLUME, ft ³ (m ³)	1010 (28.6)	894 (25.3)	982 (27.8)	841 (23.8)	1030 (29.1)
WEIGHT, ton (t)	34.9 (35.5)	33.7 (34.2)	36.8 (37.4)	33.6 (34.1)	40.7 (41.4)
VOLUME, ft ³ (m ³)	3560 (101)	3280 (92.8)	3370 (95.4)	3190 (90.3)	3420 (96.8)
WEIGHT, ton (t)	141 (143)	134 (136)	139 (141)	134 (136)	145 (147)
PUMP POWER shp (kW)	382 (285)	536 (400)	365 (272)	681 (508)	323 (241)

- WITHOUT MINIMUM WATERBOX

- WITH MINIMUM WATERBOX

TABLE 5 - CHARACTERISTICS OF OPTIMIZED COOLING SYSTEMS WITH
TYPE-1, TUBE-Y CONDENSERS OF EQUAL HEAT LOAD
(TITANIUM TUBES WITH 5/8" OD X 0.035" WALL)

MAIN CONDENSER - NO FOUNDATIONS	LENGTH, ft(m)	20 (6.10)	25 (7.63)		30 (9.15)	
	DESIGN * CONDITION	W or M	W	M	W	M
	TUBE COUNT	2940	2060	2400	1740	2452
	TUBE SW VEL., ft/s (m/s)	7.5 (2.29)	10.0 (3.05)	8.0 (2.44)	11.0 (3.36)	7.0 (2.14)
	SW FLOW gal/min (ℓ/s)	7930 (500)	7430 (469)	6920 (437)	6910 (436)	6190 (391)
	ASPECT RATIO,	2.98	4.71	4.42	6.42	5.65
	HEIGHT X WIDTH ft(m)	11.5 X 6.71 (3.51 X 2.05)	9.41 X 5.65 (2.87X 1.72)	9.59 X 5.31 (2.92X 1.62)	8.24 X 4.67 (2.51X 1.42)	8.54 X 5.31 (2.60X 1.62)
	VOLUME, ft ³ (m ³)	1080 (30.6)	890 (25.2)	981 (27.8)	838 (23.7)	1030 (29.1)
	WEIGHT, ton (t)	37.4 (38.0)	33.8 (34.3)	37.0 (37.6)	33.7 (34.2)	40.8 (41.5)
MSW SYSTEM	VOLUME, ft ³ (m ³)	3700 (105)	3320 (94.0)	3400 (96.2)	3220 (91.1)	3450 (97.6)
	WEIGHT, ton (t)	146 (148)	137 (139)	141 (143)	136 (138)	146 (148)
	PUMP POWER shp (kW)	330 (246)	549 (410)	375 (280)	714 (533)	334 (249)

* W - WITHOUT MINIMUM WATERBOX

M - WITH MINIMUM WATERBOX

TABLE 4 - CHARACTERISTICS OF OPTIMIZED COOLING SYSTEMS WITH
TYPE-1, SMOOTH-TUBE CONDENSERS OF EQUAL HEAT LOAD
(TITANIUM TUBES WITH 5/8" OD X 0.035" WALL)

LENGTH, ft(m)	20 (6.10)	25 (7.63)	30 (9.15)	
DESIGN * CONDITION	W or M	W or M	W	M
TUBE COUNT	3860	2530	1950	2400
TUBE SW VEL., ft/s (m/s)	6.5 (1.98)	9.0 (2.75)	11.0 (3.36)	8.0 (2.44)
SW FLOW gal/min (ℓ/s)	9420 (594)	8560 (540)	8030 (507)	7230 (456)
ASPECT RATIO,	2.70	4.31	6.12	5.66
HEIGHT X WIDTH ft(m)	12.0 X 7.41 (3.66 X 2.26)	9.66 X 5.80 (2.95 X 1.77)	8.34 X 4.90 (2.54 X 1.49)	8.55 X 5.30 (2.61 X 1.62)
VOLUME, ft ³ (m ³)	1280 (36.2)	1030 (29.1)	903 (25.6)	1030 (29.1)
WEIGHT, ton (t)	44.3 (45.0)	38.7 (39.3)	36.2 (36.8)	40.8 (41.5)
VOLUME, ft ³ (m ³)	4530 (128)	3770 (107)	3450 (97.6)	3550 (100)
WEIGHT, ton (t)	177 (180)	154 (156)	144 (146)	150 (152)
PUMP POWER shp (kW)	197 (147)	236 (176)	290 (216)	206 (154)

* W - WITHOUT MINIMUM WATERBOX

M - WITH MINIMUM WATERBOX

Tables 4-7. For Type-1-condenser steam conditions, the following trends are noted.

- (1) For equal-length condensers, coolant flow rate varies inversely with the degree of enhancement.
- (2) Except for the TFCP, tube flow velocity (and, therefore, overall U) in the enhanced case shows less dependence on condenser length than does the smooth case.
- (3) By applying the data of Tables 4-7, it can be further shown that, for a given enhancement,
 - (a) coolant flow rate decreases with increasing condenser length
 - (b) coolant velocity varies directly with condenser length
 - (c) $\Delta T \ell_m$, the log mean temperature difference, varies directly with coolant flow rate (and, therefore, inversely with length) and becomes more sensitive to flow as flow decreases.
 - (d) the dependence of loop weight (excluding condenser) on coolant flow diminishes as the flow falls below 8,000 gal/min (505 l/s)
 - (e) the influence of condenser length on pump weight is much greater for an enhanced system

Moreover, items (1) and (3c) may be combined to state the following:

- ° For equal-length condensers, $\Delta T \ell_m$ varies inversely with the degree of enhancement
- ° For equal changes in condenser length, the fractional change in $\Delta T \ell_m$ varies directly with the degree of enhancement.

An important implication in the above statements is that as condenser length increases, the higher U and lower loop weight of the enhanced system are severely offset by the lower $\Delta T \ell_m$ and higher pump weight. A further implication is that the performance of Type-2 systems may not have the same dependence on condenser aspect ratio as Type-1 designs.

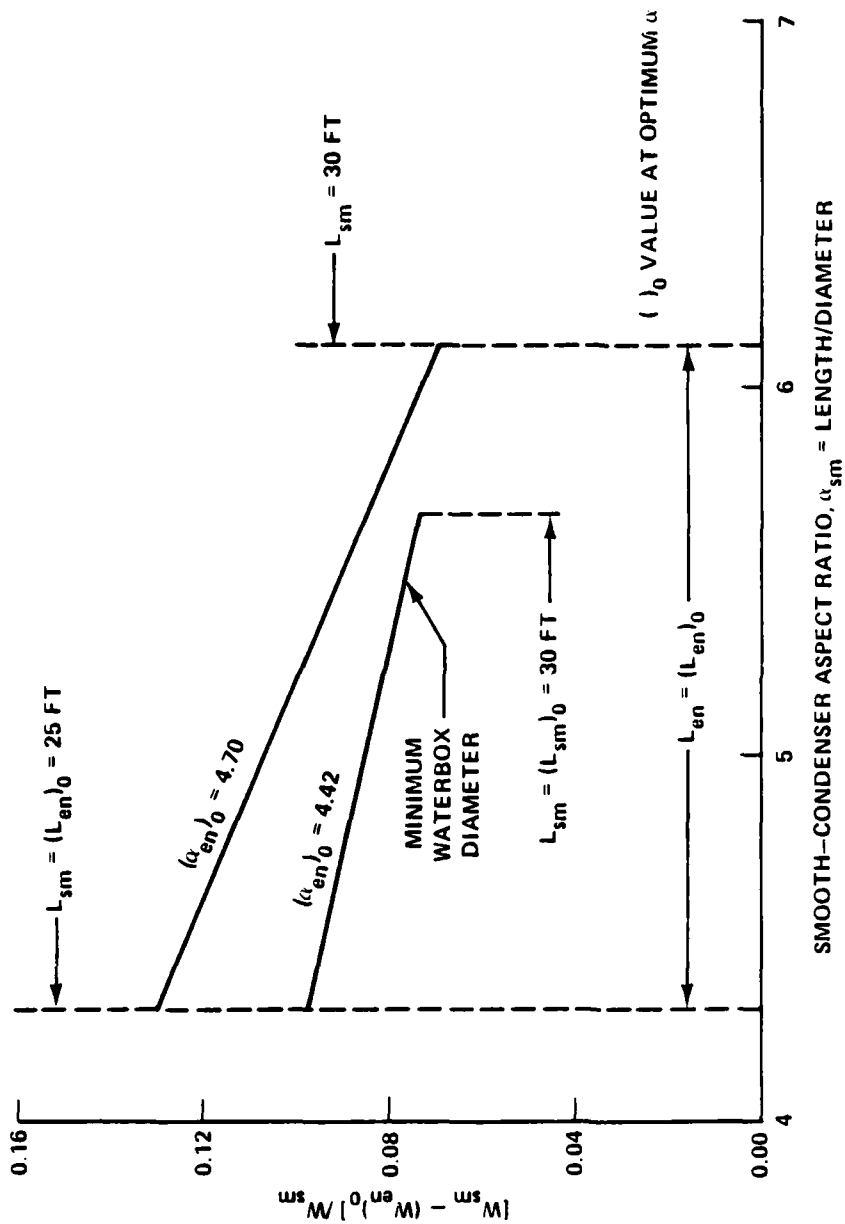


Figure 12 - Effect of Variable-Length Smooth Condensers on the Cooling-System Weight Benefit of Optimum-Aspect-Ratio, Tube-YS Condensers (Type-1 Designs)

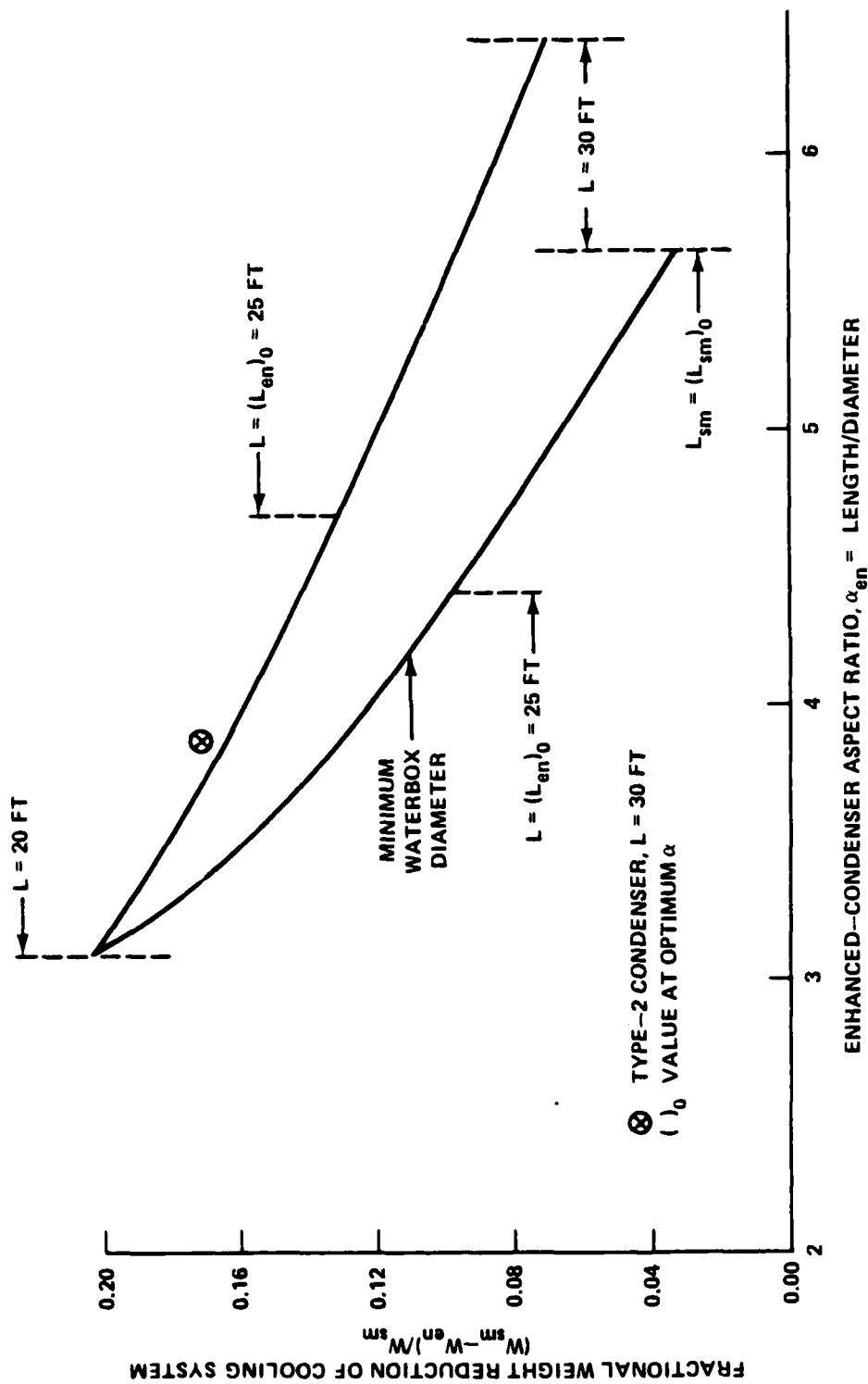


Figure 11 - Variation of Cooling-System Weight Benefit with the Aspect Ratio of a Tube -YS Condenser (Equal-Length, Type-1 Designs)

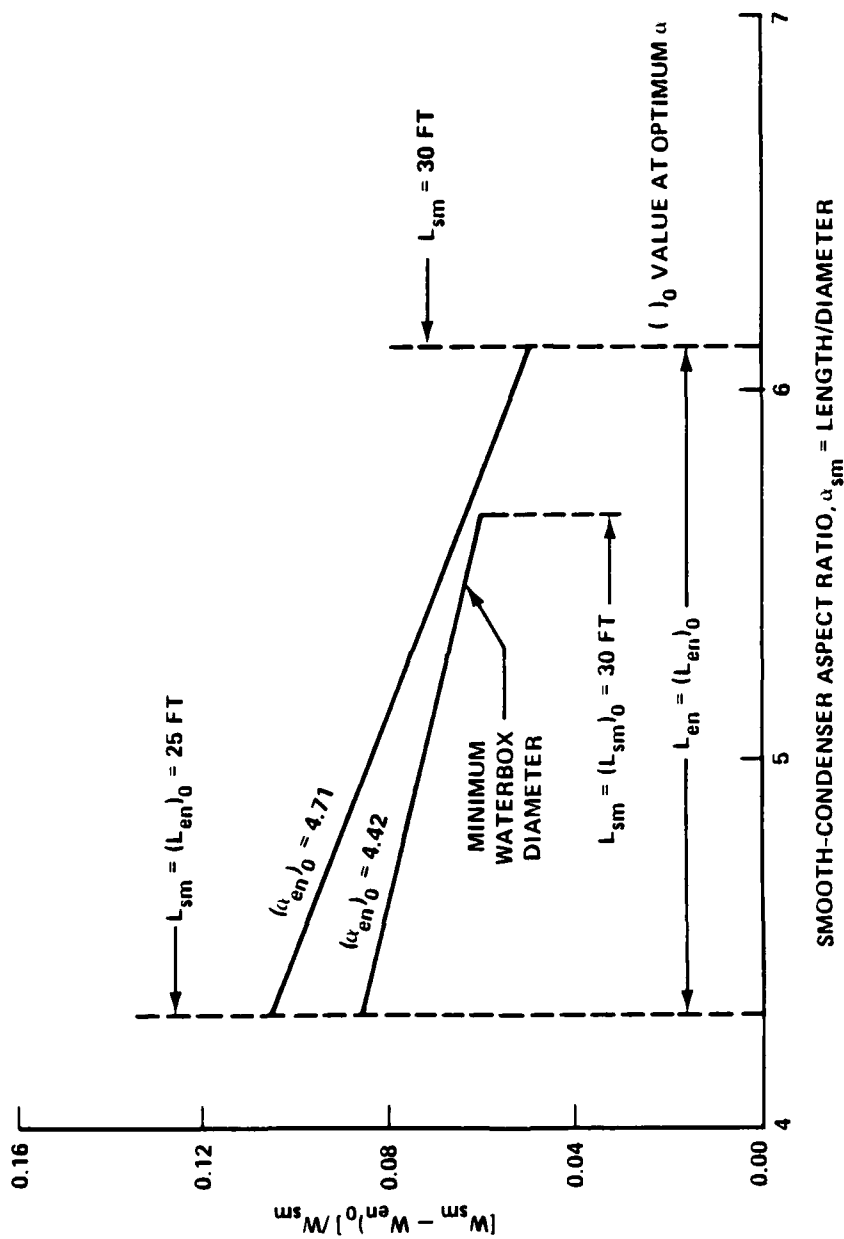


Figure 10 - Effect of Variable-length Smooth Condensers on the Cooling-System Weight Benefit of Optimum-Aspect-Ratio, Tube-Y Condensers (Type-1 Designs)

The effect of condenser vacuum on weight benefit was examined for a single design case. This is represented by the 30-ft (9.15 m). Type-2 condenser with a pressure of 2.5 in. Hg (8.45 kPa). It is seen that for the specified length, the pressure change simply reduces the optimum aspect ratio from a value of around 6 to a value of 3.7. Therefore, for a given condenser length, enhancement benefit increases as the condenser pressure decreases (see Figure 9).

Figure 10 shows how the weight benefit can be altered by "freezing" the enhanced condenser at its optimum α ($L = 25$ ft (7.62 m)). It is seen that while the unconstrained-condenser benefit has changed little, the constrained-condenser benefit at a smooth length of 30 ft is definitely improved. For the constrained case, no further reduction in benefit is expected for lengths beyond 30 ft (9.15 m).

The effect of applying tube YS, rather than tube Y, to the condenser is shown in Figure 11. For low-aspect-ratio designs, weight benefits of about 20 percent can be attained. However, as α is increased, the divergence between the constrained and unconstrained designs is larger than for tube Y. The Type-2 performance of 13 percent for tube Y is now increased to nearly 16 percent.

Figure 12 shows trends that are very similar to those of Figure 10. As expected, the curves are displaced upward, and the benefit of applying the optimum- α enhanced condenser to the constrained case is again clearly evident. It should be noted that for the cases considered, the degree of enhancement has little effect on the optimum aspect ratio or size of an enhanced condenser in a flow-optimized system.

A summary of the important system characteristics for each of the condenser geometries and lengths evaluated in the Type-1 and Type-2 designs is given in

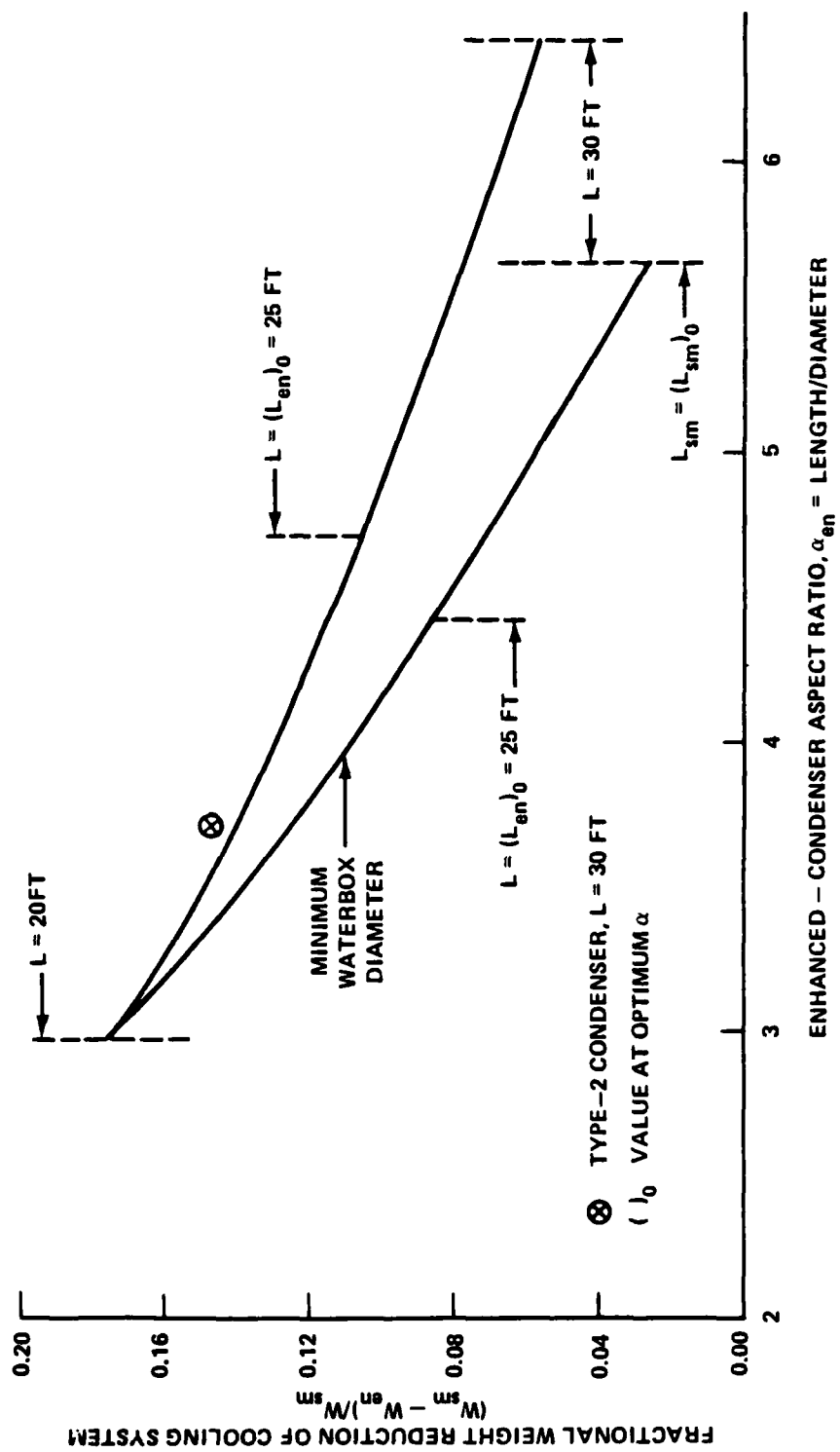


Figure 9 - Variation of Cooling-System Weight Benefit with the Aspect Ratio of a Tube -Y Condenser (Equal-Length, Type-1 Designs)

unacceptable friction penalty. The result is a tube with fairly well balanced internal and external resistances. It should be noted that fouling resistance was the same for all three tubes in Figure 8.

CONDENSER LENGTH

Figure 9 demonstrates the effect of condenser length (or aspect ratio α) on system weight benefit when tube Y is employed in a Type-1 condenser. The upper curve represents the performance with no design constraints (other than the tube vendor's estimates for o.d. sinkage, a result of the enhancement process). However, one important constraint does exist where long condensers are desired for system minimization. In such cases, considerations related to flow headering and maintenance access impose a definite lower limit on the waterbox diameter. Consequently, a minimum-waterbox condition was imposed on the optimization approach. With this constraint, the lower curve of Figure 9 could only be flow-optimized for aspect ratios below that corresponding to the waterbox limit. At higher aspect ratios, the weight was determined by the minimum-waterbox condition rather than the optimum-waterbox condition. Since the 50-in. (127-cm) minimum chosen for the waterbox diameter doesn't affect low-aspect-ratio condensers, the curves diverge from a common vertex near an aspect ratio of 3. In general, aspect ratio has a profound effect on weight benefit, and, for an unconstrained waterbox, a change in aspect ratio from 3 to 6.4 causes a decrease in weight benefit from 17.5 percent to about 5.5 percent. From a review of the data, it was learned that enhanced-condenser lengths above 25 ft and smooth-condenser lengths above 30 ft have a negligible effect on system weight. The aspect ratio was, therefore, taken to be optimum at these points; these optimum points are indicated on the curves as $L = (L_{en})_o$ or $L = (L_{sm})_o$.

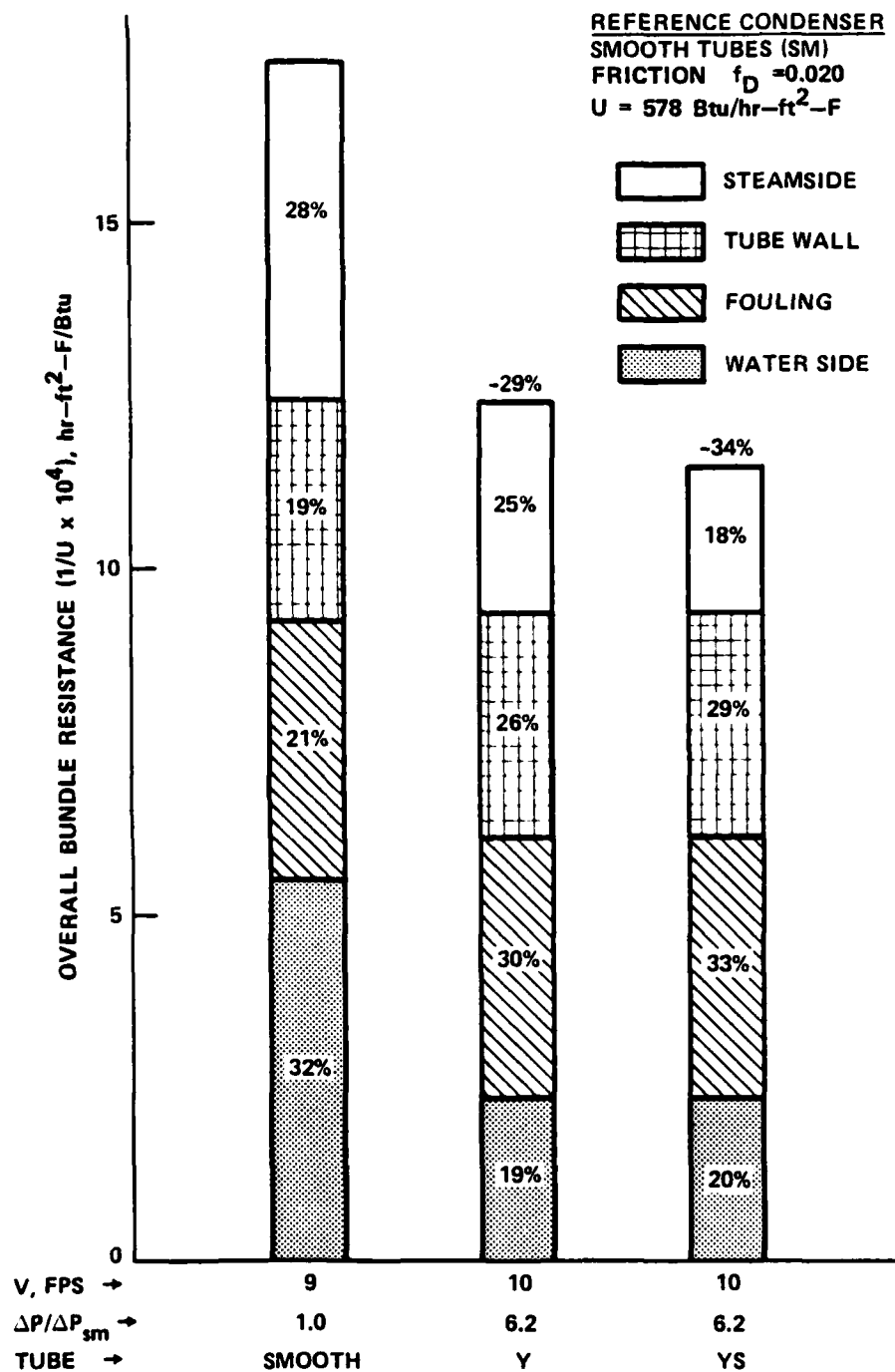


Figure 8 - The Effect of Tubes Y and YS on the Thermal Resistance of 25-Foot, Type-1 Main Condensers in Optimized Cooling Systems (85%-Clean Titanium Tubes, 5/8" O.D. x 0.035" Wall)

and seven percent larger than a system derived from overall-minimization techniques. In addition, a dramatic 38-percent increase in system pump power is required for the minimum-condenser design.

The observed minimum in the system curve of Figure 7 is attributed to the manner in which the loop weight (condenser excluded) varies with the flow. Below 9700 gal/min (612 ℓ/s), loop weight is a weaker function of flow than condenser weight, which then dominates the overall system weight. However, as the condenser weight reduction tapers off at flows above 9700 gal/min, the expanding diameter of the loop piping causes loop weight to become more flow sensitive. Hence, for flows above 9700 gal/min, loop weight is the dominant factor in system weight. In view of these observations, the minimum-system (flow-optimization) approach was universally applied to the present study.

TUBE WALL AND FOULING RESISTANCE

A breakdown of the various resistances controlling the tube heat transfer in the three tube geometries is shown in Figure 8. The comparison is made for 25-ft (7.62-m), Type-1 condensers operating at optimum-system flow conditions. In the case of tube Y, the overall resistance has been reduced by 29 percent, and the waterside and steamside resistances no longer dominate the overall resistance. However, the bundle pressure loss has increased to six times that in a smooth bundle. Tube YS further reduces the steamside resistance to the point where it accounts for only 18 percent of the overall resistance. The relative magnitudes of the resistances in this tube clearly illustrate the inherent limitations imposed on enhancement when the combined tube wall and fouling resistance becomes dominant. Beyond this point, only very large increases in either waterside or steamside enhancement will be able to effect any significant reductions in overall resistance. In developing tube YS, further waterside enhancement was not viable because of the

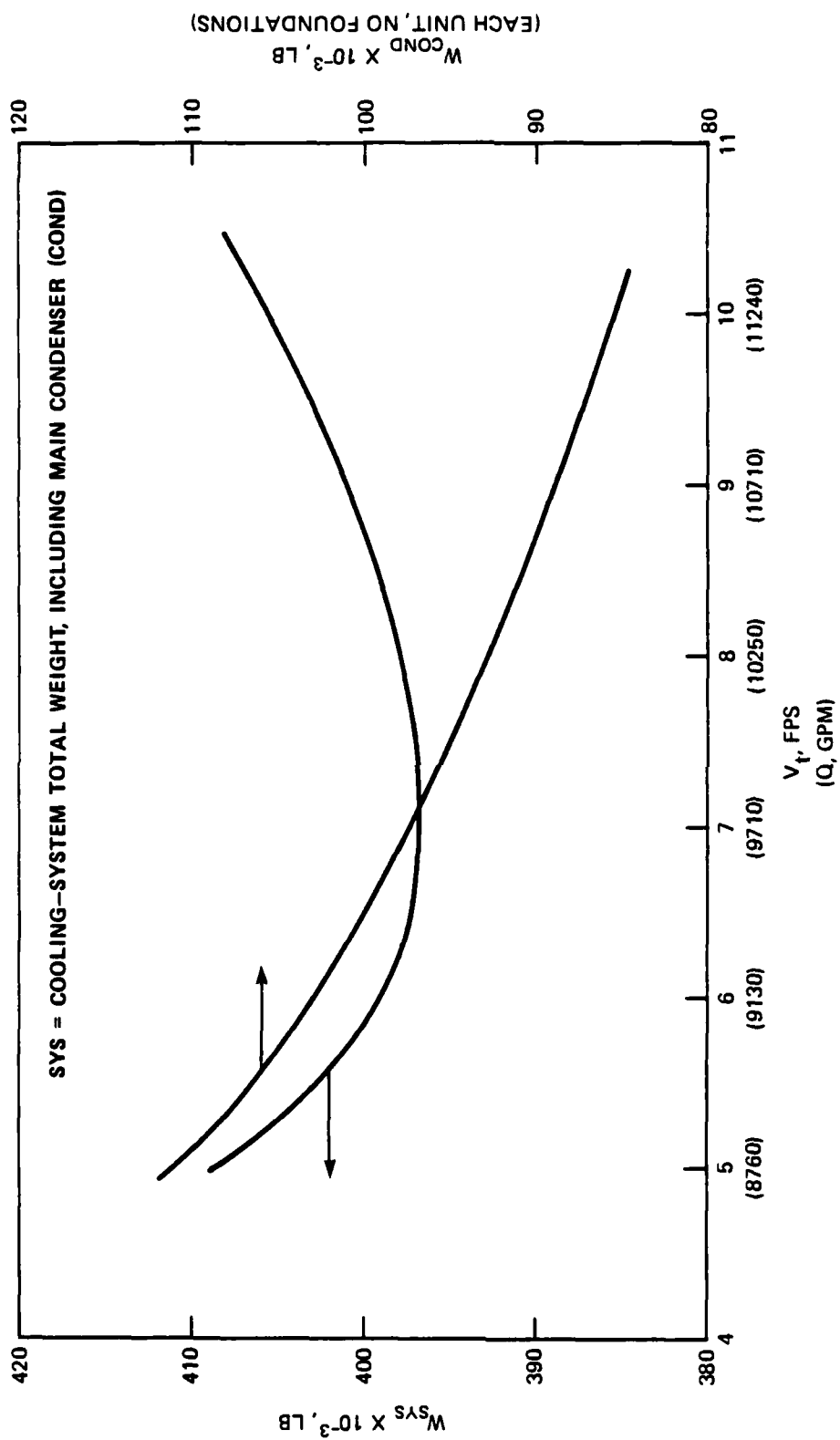


Figure 7 - Variation of System Weight and Condenser Weight with Condenser-Tube Flow Velocity (20-Ft, Type-1 Condenser with Smooth Titanium Tubes)

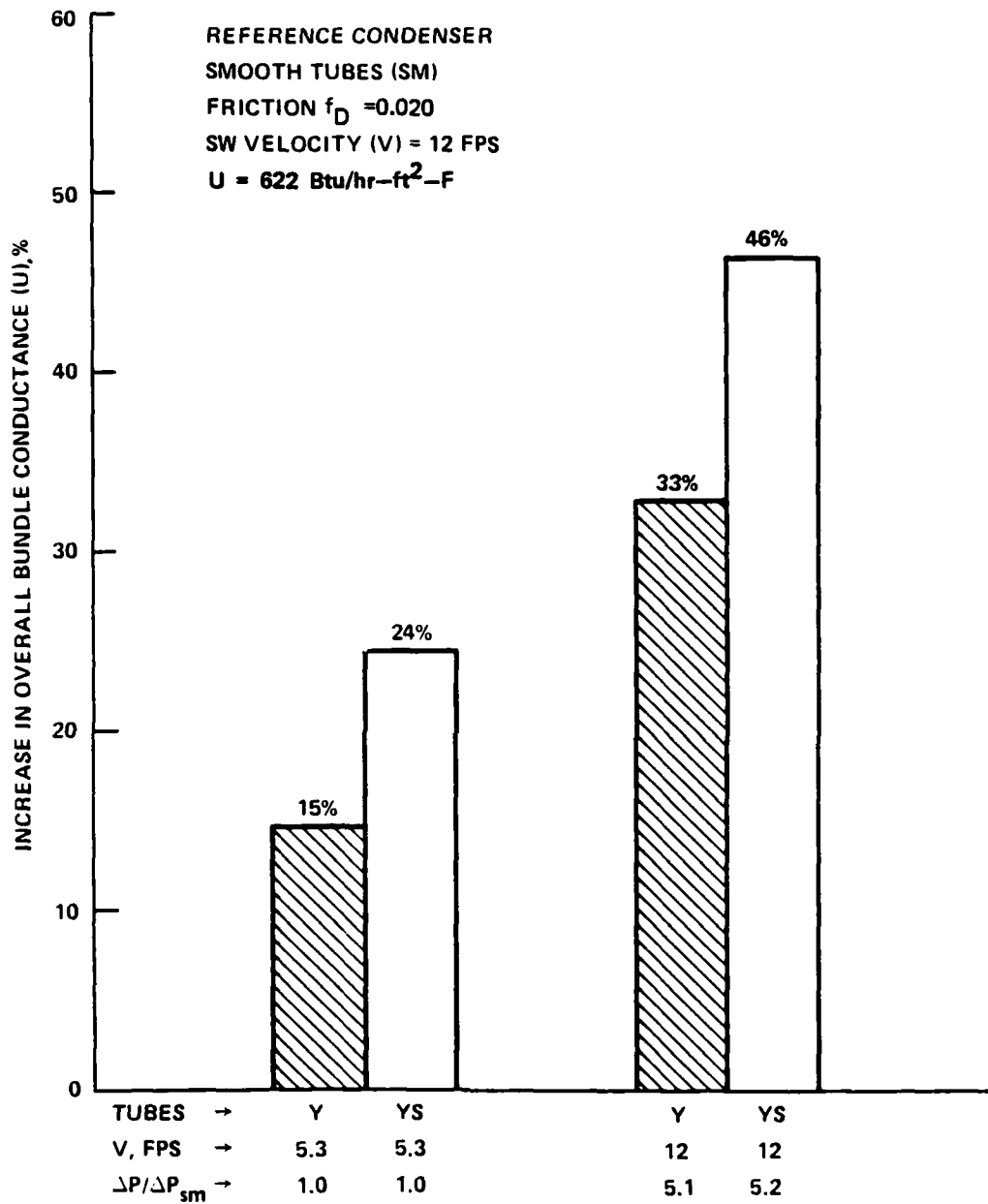


Figure 6 - Effect of Tubes Y and YS on the Overall Bundle Conductance of a 25-Foot, Type-1 Condenser (85%-Clean Titanium Tubes, 5/8" O.D. x 0.035" Wall)

TABLE 7 - CHARACTERISTICS OF OPTIMIZED COOLING SYSTEMS WITH
30-FOOT, TYPE-2 CONDENSERS OF EQUAL HEAT LOAD
(TITANIUM TUBES WITH 5/8" OD X 0.035" WALL)

MAIN CONDENSER - NO FOUNDATIONS	TUBE	SMOOTH	Y	YS
	TUBE COUNT	4480	4000	3600
	TUBE SW VEL., ft/s (m/s)	8.5 (2.59)	8.5 (2.59)	9.5 (2.90)
	SW FLOW gal/min (ℓ/s)	14300 (902)	12300 (776)	12300 (776)
	ASPECT RATIO,	3.61	3.72	3.88
	HEIGHT X WIDTH ft(m)	10.5 X 8.30 (3.20X 2.53)	10.3 X 8.07 (3.14X 2.46)	10.1 X 7.76 (3.08X 2.37)
	VOLUME, ft ³ (m ³)	2250 (63.7)	2140 (60.6)	2000 (56.6)
	WEIGHT, ton (t)	59.7 (60.7)	55.7 (56.6)	52.5 (53.3)
MSW SYSTEM	VOLUME, ft ³ (m ³)	9050 (256)	7730 (219)	7490 (212)
	WEIGHT, ton (t)	277 (281)	242 (246)	236 (240)
	PUMP POWER shp (kW)	330 (246)	759 (566)	916 (683)

Tables 4-7 also indicate the effect of condenser length on system pump power. While the pump power of the minimum-waterbox designs shows little dependence on length, the unconstrained designs show that a length increase from 20 to 30 ft (6.10 - 9.15m) causes a dramatic increase in tube flow velocity. This causes pump power to increase by 47, 116, and 78 percent for smooth, Y, and YS tubes, respectively. Moreover, a Type-2 condenser using tube Y needs only a slightly higher pump power than an equal-length, unconstrained Type-1 design. However, on the basis of equal aspect ratio, the Type-2 designs, which operate at much higher flows, require much greater pump power.

ENHANCED-TUBE O.D. SINK

In examining the effect of enhanced-tube o.d. sink (due to the roping process) on system performance, all of the data for Type-1 condensers, with both tubes Y and YS, were evaluated with and without tube-sink effects. Although there did appear to be a detectable effect in some of the longer condensers, its magnitude appeared to be nearly within the accuracy of the calculation procedure. It was, therefore, reasonable to conclude that the sinkage predicted for these tubes has an insignificant effect on the weight benefits due to enhancement. However, close examination of some high-aspect-ratio data clearly indicated that for coolant flows below 6500 gal/min (410 ℓ/s), the cooling system performance is markedly reduced by sink effects. In general, such flows are only realized in high-aspect-ratio condenser design, and as length increases, the effects will first be encountered in designs with a waterbox constraint. For the tube size examined, the initial encounter is at a length of about 28 ft (8.54 m). Since the sink effect will increase as the flow decreases, the weight benefit for equal-length designs will continue

to fall even after the smooth-tube optimum length of 30 ft (9.15 m) has been reached. On the basis of equal o.d. sinkage for both 5/8 and 9/16 in. (1.59 and 1.43 cm) tubes, it can be shown that for the latter size, the length at which the critical flow is encountered is at least 2 ft (0.610 m) greater than for 5/8 tube. This is a direct result of the higher flows associated with smaller tube diameters.

PUMPING POWER REQUIREMENTS

From an examination of condenser-tube flow velocity and system pumping power, it is possible to show that, for a given set of system operating temperatures and pressures, a unique relationship exists between power and velocity for each tube-surface enhancement. The relationships found for the present geometries are shown graphically in Figure 13; they show a very weak dependence on condenser length, with the maximum deviation from the average being only 7.5 percent. Consequently, for given operating conditions, the system pumping power for a given enhancement geometry may, for all practical purposes, be assumed to be solely dependent on the velocity in the condenser tubes. This fact greatly facilitates system comparisons at different coolant velocities.

CONCLUSIONS

SUMMARY

This design study has demonstrated the effective use of an analytical model to minimize the cooling-water-system weight and volume associated with high-performance marine power condensers. The model can accommodate a wide range of variables, including the effects of steam dynamics (vapor velocity and inundation effects) and tube enhancement. As a result of this detailed simulation, it has been possible to size condensers for a wide variety of Naval applications and operating requirements. DTNSRDC expanded the detailed

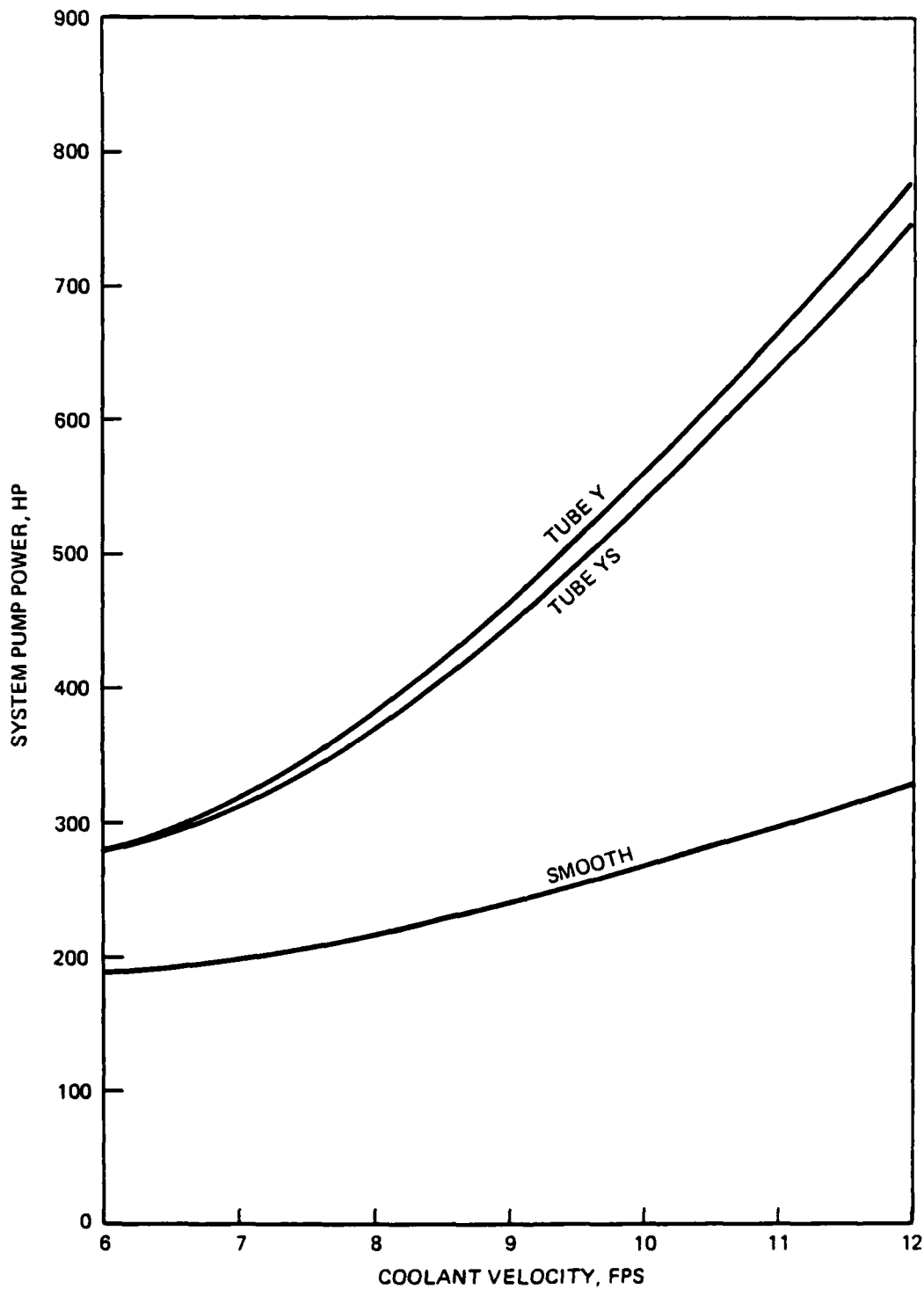


Figure 13- Variation of Cooling-System Pumping Power with Tube Flow Velocity in a Type-1 Condenser (Mean Values for 5/8-inch-O.D. Tubes in a Condenser Length Range of 20-30 Feet)

tube-bundle model provided by Nunn and Marto¹⁵ to permit evaluation of the overall condenser configuration (shell, waterboxes, etc.). This model was then incorporated into still another computer code capable of sizing an entire main cooling system. A comparison of condenser modeling results with those for an existing operational unit showed size differences of less than 4 percent (see Table 2).

In conformance with new design trends, the present study utilizes lightweight, high-strength titanium and inconel materials in all system components (titanium tubes, inconel waterboxes and piping, etc.). Performance evaluations were conducted for three condenser-tube geometries--a smooth tube and two types of enhanced tubes. The tubes had a 5/8-in. (1.59 cm) o.d. and 0.035 in. (0.089 cm) wall and were assumed to be 85-percent clean (fouling resistance of 3.3×10^{-4} h-ft²-F/BTU (5.81×10^{-5} m²-C/W)). The overall performance of these condenser-tube bundles is shown in Figure 8; it reflects the limiting effect of the thermal resistance due to the tube wall and fouling as the internal and external convective resistances are reduced by enhancement. Additional improvement in heat transfer can only be achieved by reducing the tube wall thickness or fouling resistance. The spirally grooved (roped) tube* shown in Figure 3 is a low risk, commercially available surface enhancement suitable for titanium-tube condensers in shipboard cooling systems. While the steamside inundation effects of these tubes are comparable to those for smooth tubes, they can be virtually eliminated by the TFCP (Figure 4), which has commercially demonstrated performance gains. For Type-1 systems, weight benefits with tube Y can reach 17.5 percent for a condenser aspect ratio (length divided by diameter) of about 3.0. With the TFCP, the benefit is increased by another 2.5 percentage points.

* As a result of titanium-tube biofouling and manufacturing limitations, other surface enhancements were considered high cost, and high risk developments.

EFFECT OF CONDENSER DESIGN ON ENHANCED-TUBE BENEFITS

It has been shown that because of the opposing influences of coolant flow rate on condenser weight and cooling-loop weight, flow optimization is necessary if minimum system weight or volume is to be achieved. The variation of system weight benefit with condenser aspect ratio was determined by flow optimization at various condenser lengths between 20 and 30 ft (6.10 and 9.15 m). The results show that equal-length dimensionally unconstrained Type-1 condensers with tube Y yield system weight benefits from 17.5 to 5.5 percent for aspect ratios from 3 to 6.4, respectively. When the equal-length approach was applied to condensers with a minimum waterbox diameter of 50 in. (19.7 cm), the deviation from the unconstrained design case increased markedly with aspect ratio. Moreover, the present study indicates that the performance of the enhanced condensers was optimum at aspect ratios of 4.71 and 4.42 for the unconstrained and constrained cases, respectively. The selection of aspect ratios greater than the optimum values adversely affects either pump power (unconstrained case) or weight benefit (constrained case).

The results for tube YS were analogous to those for tube Y, with the unconstrained performance lying 2.5 percentage points above those for tube Y. Although the divergence between the unconstrained and constrained cases was larger than for tube Y, the use of the optimum-aspect-ratio enhanced design again proved to be an effective remedy for this effect. The optimum aspect ratios for tube-YS condensers are the same as those found for tube Y.

In Type-2 condensers using either tube Y or YS, the benefit correlated well with the aspect ratios of the Type-1 designs. Tube YS again yielded a benefit about 2.5 percentage points above that for Tube Y. These results clearly show how high condenser vacuum favors the use of enhancement. Moreover, the benefits of Type-2 condensers at higher aspect ratios may exceed those of Type-1 designs.

For the unconstrained design case, pump power increases markedly with condenser aspect ratio; the specific increases over the length range were 47, 116, and 78 percent for smooth, Y, and YS tubes, respectively. While the pump power of a Type-2 system differs little from that for a system with an unconstrained Type-1 condenser of equal length, it is markedly above that for a Type-1 of equal aspect ratio. Moreover, for systems in which all condenser conditions except coolant flow rate are fixed, tube flow velocity may be used to characterize the system pumping power for a given tube geometry.

A thorough assessment of the effects of o.d. sink on enhanced-tube performance indicated that such effects, though they may exist at high aspect ratios, are within the overall calculation accuracy. Sinkage effects were, therefore, assumed negligible and had no impact on the final results.

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APPENDIX
STEAM CONDENSER
THIN-FILM-CONDENSATE PROMOTER*

INTRODUCTION

In recognizing the need for efficient heat-transfer surfaces to yield smaller size and, thereby, less costly heat exchangers, Union Carbide has developed, over the past several years, numerous boiling and condensing enhancements for commercial applications. One such condensing augmentation is the Thin Film Condensing Promoter (T.F.C.P.) [Invention Disclosure 863 347 (Reference A-1)] which can readily be incorporated externally to tubing of varying diameters and materials. The unique feature of this enhancement is that the tube's heat-transfer performance is unaffected by the extent of condensate loading. The condensing coefficient of a plain tube is degraded when condensate "rain" from overhead tubing "floods" the tube surface causing an additional liquid layer which increases thermal resistance.

General Description of T.F.C.P.

The Thin Film Condensing Promoter (T.F.C.P.) is basically a wire strand which is firmly and spirally wrapped around the tubing's outside surface. The wire diameter is typically 1/16 in. (0.159 cm) and the coil-to-coil spacing (pitch) 0.3 in. (0.762 cm). These dimensions were established from theoretical and experimental optimization studies.

The Thin Film Condensing Promoter is essentially a liquid-surface-tension device which rapidly "thins out" and "strips" the condensate layer on the outside tube surface, thereby significantly improving the condensing heat transfer

* Condensed from Reference 10 in main text.

ver loaded plain tubing. If we consider a tube in a large steam-condenser bundle, the condensate formed on the tube, plus the condensate accumulated from tubes above, is quickly drawn, by surface-tension forces, to the interface formed between the T.F.C.P. and the tube itself, thus allowing only a very thin layer of condensate between two adjacent strands of the T.F.C.P. The thinning of the condensate layer decreases the condensing thermal resistance and thereby increases the heat-transfer coefficient by a factor of 2-3 times over that of a loaded bare tube.

Demonstration Experiments

The effectiveness of the T.F.C.P. has been demonstrated for various applications in single-tube tests (with "loaded" conditions simulating bundle effects) condensing steam (under vacuum and pressure), ammonia, R-12 and propylene. Tube materials were aluminum, copper-nickel, and copper. Test data generally showed condensing performance that was 2-3 times better than a loaded bare tube.

In addition, representative tube-bundle experiments (one bundle with, the other without, the T.F.C.P.) were conducted in Union Carbide's Steam Condensing Test Facility. The bundle contained 84 horizontal copper/nickel tubes (in columnwise arrangements of 24 tubes aligned directly above and below each other). The condensing-heat-transfer measurements were actually higher than determined from the loaded single-tube tests. The condensing performance data presented below are based on loaded and unloaded single-tube tests.

In addition, for the OTEC* program (under DOE contract), Union Carbide designed and built an ammonia condenser (3 MW_{th} heat duty) which used the

* Ocean Thermal Energy Conversion

T.F.C.P. enhancement. This heat exchanger was successfully tested by Argonne National Laboratories. Experimental results were somewhat better than was predicted by the single-tube tests. Recently, Union Carbide supplied, under an E.P.R.I.* contract, a steam condenser (60 MW_{th} heat duty) employing the T.F.C.P.; since it serves as a standby unit in the cooling system of a 10-MW electrical power station in Bakersfield, California, a specific operating date has not been given.

OPTIMUM T.F.C.P. CONFIGURATION

During recent studies at Union Carbide Corporation, an experimental/theoretical analysis has been conducted to arrive at an optimum wire diameter and pitch combination for typical tube diameters that are condensing steam under vacuum. The principal criteria and the results were:

(1) The wire diameter had to be greater than the radius of curvature of the condensate rivulet, otherwise flooding will occur. It was found that a wire diameter of 0.055 in. (0.140 cm.) is required.

(2) The optimum pitch between two adjacent strands of wire was determined to be 0.3 in. (0.762 cm.).

Consequently, a tube sample was prepared to meet those criteria.

DESCRIPTION OF T.F.C.P. - ENHANCED-TUBE SAMPLE

The attached photograph (Figure A-1) shows the finished sample tube, which is 6 inches long, with a 0.061 in. (0.155 cm.) diameter titanium wire wound tightly in the spiraling grooves. While it is not necessarily a requirement that the wire be tight against the tube for good condensation, it was considered that for this application, a tight interference between the wire and tube would be essential. To achieve this requirement, it was necessary to pre-wind the

* Electric Power Research Institute

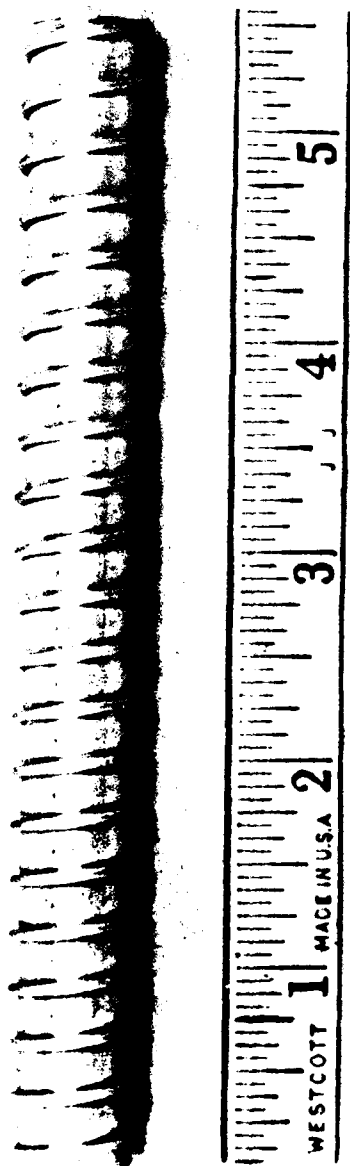


Figure A-1 - Spiral-Ribbed Tube Sample with Union Carbide T.F.C.P.

wire onto a mandrel with a diameter somewhat less than the groove diameter of 0.589 in. (1.50 cm.). The wire was first wound onto this mandrel in a close-coiled fashion by securing the start against the mandrel and continuously rotating until sufficient turns were produced. On releasing the wire, spring-back occurred, giving a spring I.D. of approximately 0.567 in. (1.44 cm). The spring was then further expanded lengthwise until the pitch between turns was slightly greater than 0.300 in. (0.762 cm); when released, it yielded a pitch close to 0.300 in. (0.762 cm). By rotating the tube in a clockwise direction, the spring was readily wound into the grooves on the tube. By this method, the wire had an interference fit against the tube, fulfilling the requirement for a tight T.F.C.P. Cutting off excess wire and smoothing up the ends finished the sample.

POTENTIAL PROCESS METHODS FOR MASS PRODUCING T.F.C.P. - ENHANCED TUBING

The technique used to apply the T.F.C.P. to a sample tube (6 in. (15.2 cm.) long) would not be suitable for large-scale production on long tubes with plain areas (not corrugated) located at the tube supports. The problems with the present installation technique include the following:

- (1) The processes for preforming the wire and manually installing it are too slow and costly. Also, the friction between the wire and the tube increases with tube length, thus making it progressively more difficult to thread the wire coil on the tube corrugations.

- (2) The plain areas of the tube at the tube sheet and tube supports are 0.625 in. (1.59 cm) in diameter, which is larger than the diameter of the corrugation (0.589 in. (1.50 cm)). Threading the precoiled T.F.C.P. over the bare areas of the tube requires that the coil be expanded, thus creating excessive drag during installation.

The application process to be used for production must provide the following:

(1) The T.F.C.P. strand must be tightly applied to the tube to preclude installation problems when the tube is laced through the tube supports and to prevent any vibration (noise) between the T.F.C.P. and the tube. As previously discussed, it is not necessary for the T.F.C.P. to be tight on the tube to achieve the condensing enhancement.

(2) The uncorrugated areas of the tube at the tube-support location must have a diameter that is equal to or slightly greater than the diameter of the T.F.C.P. so that the clearance between the hole in the tube support and the tube meets code requirements.

(3) The T.F.C.P. wire material and attachment method must not cause any potential degradation of the titanium tube due to corrosion or other metallurgical phenomena.

To achieve adequate production speed and T.F.C.P. tightness on the tube, it is felt that the T.F.C.P. should be coiled directly on to the tube in the valley of the corrugation. The most promising approach for the bare tube areas at the tube-support locations is to close-coil the T.F.C.P. so that the strands are touching each other, thus producing a diameter of $5/8 + 1/16 + 1/16 = 3/4$ in. (1.91 cm). It is believed that if the T.F.C.P. wire is heated in an inert atmosphere prior to winding, the residual stress caused by the coiling will be reduced, which will result in less spring-back of the coil than would occur if the wire were wound on the tube at room temperature. In addition, the thermal contraction of the hot wire should provide sufficient shrinkage to assure that the T.F.C.P. is tight on the tube. The tube would not be heated during the application of the T.F.C.P. If a 304 stainless-steel wire were used, the thermal contraction would be about twice that achieved with a titanium wire

heated initially to the same temperature. This results from the difference in thermal-expansion coefficients for the two materials. If the winding of the heated T.F.C.P. wire does not achieve the desired tightness, it will be necessary to consider other restraint aids such as tack welding the wire directly to the tube, followed by winding and another weld at the finish. Metallurgical evaluation would be required to assure that the weld does not compromise the integrity or mechanical properties of the titanium tube.

In the unlikely event that close winding at the tube-support locations proves to have mechanical problems, it would be possible to install split sleeves at these locations, with an outer diameter that is equal to or greater than the diameter of the T.F.C.P.

Detailed investigation of the above proposed installation methods will be needed to verify their feasibility.

STEAM CONDENSING COEFFICIENT ESTIMATE

The condensing heat-transfer performance (heat flux as a function of condensing temperature difference*) for plain (unenhanced) tubing, predicted by the well-known Nusselt theory, is shown on Figure A-2 by the dashed lines. One line represents the unloaded or single-tube condition, and the other represents a tube loaded 20 times with condensate, simulating the 20th tube down in a vertical row of a large bundle. Also shown in Figure A-2 is the condensing-heat-transfer performance of the T.F.C.P. - enhanced tube. It was observed from experimental work that the single T.F.C.P. tube and the loaded T.F.C.P. tube yielded identical condensing heat transfer. The T.F.C.P. performance is based on several experimental tests conducted in steam at 4.5

* ΔT_c = Steam Bulk Temperature - Outside Tube Wall Temperature

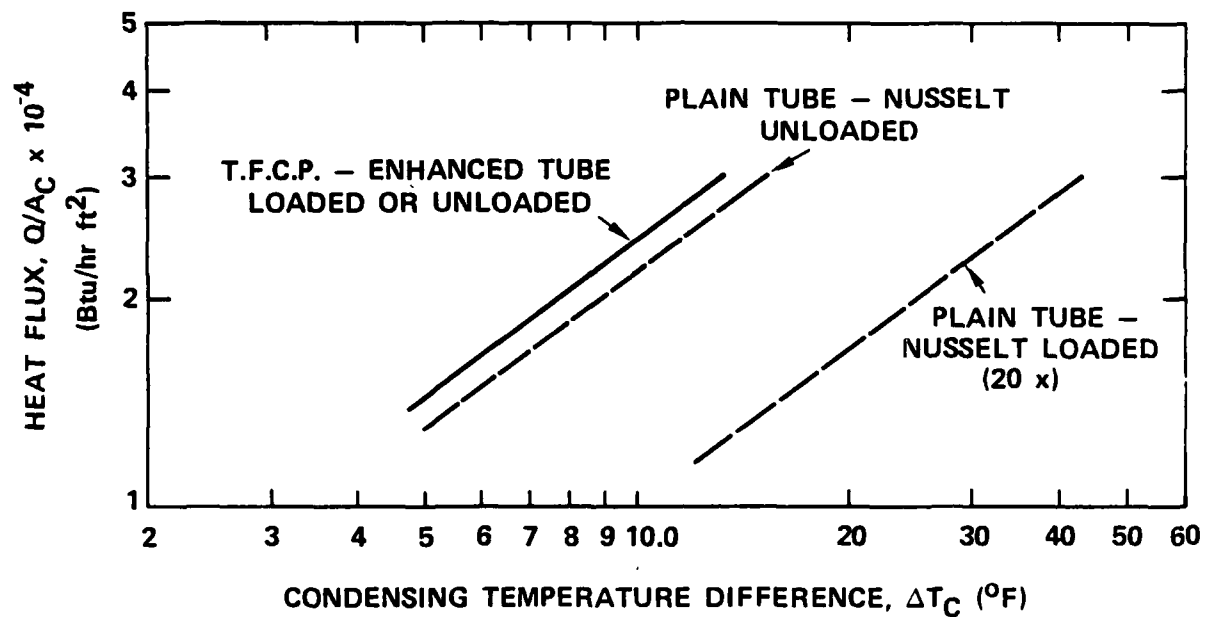


Figure A-2 - Heat Flux vs Condensing Temperature Difference: Horizontal Steam Condensation on Union Carbide TFCP-Enhanced Tube at 4.5" HgA

in. HgA (15.2 kPa). The plain single-tube heat transfer, under the same unloaded conditions, resulted in about the same performance as the T.F.C.P. As indicated above, the enhancement by the T.F.C.P. tube is achieved by the rapid stripping of the condensate, which prevents the usual performance degradation associated with plain-tube loading. A typical result shown in Figure A-2 is that at a heat flux of 30,000 Btu/hr-ft² (94.5 kW/m²), the condensing temperature difference for the T.F.C.P. is 13 F (7.2 C), while that of a predicted plain tube, loaded 20 times, is 42 F (23 C); thus an enhancement over 3 times is achieved. The loaded plain tube condensing performance was found to be slightly better than the Nusselt prediction.

Figure A-3 shows the ratio of condensing coefficient of the nth tube in a vertical column to that of the top tube as a function of the number of tubes in the column (assuming that the horizontal tube rows are arranged in an in-line pattern as shown). For the T.F.C.P., the condensing heat-transfer coefficient remains constant regardless of the number of tubes in the vertical column (or loading) in a bundle. The figure also shows the condensing performance of a plain tube versus the number of tubes in the vertical column; it includes the predictions of Nusselt, Kern, and Withers and Young (Reference A-2). Since the bundle designs which are appropriate to naval application will probably have no more than 30 tubes in the column, the evaluation of flooding effects was based on 30 in-line tube rows. For that case, the bottom-row tubes will exhibit a condensing coefficient 0.32 to 0.47 that of the top tube, while in a T.F.C.P. enhanced bundle, the condensing performance of the 30th row is unchanged.

Figure A-4, the integrated form of Figure A-3, represents the ratio of the average bundle condensing coefficient to that of the top tube as a function of the number of tubes in the vertical column. Again, the trends are the same as explained in Figure A-3, with the T.F.C.P. being unaffected by condensate loading of the tube.

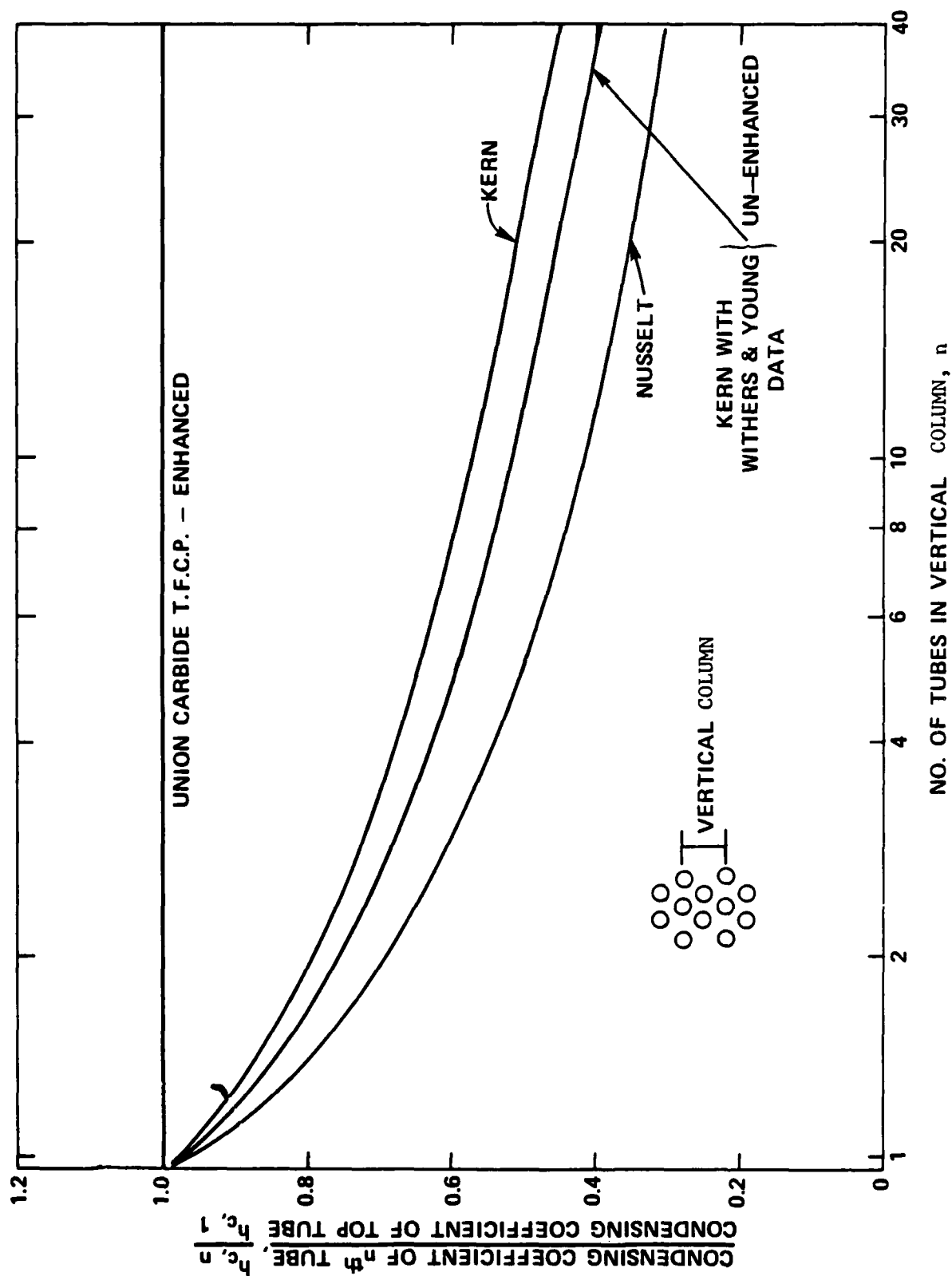


Figure A-3 - Correction to Steam Condensing Coefficient due to Condensate Loading on n^{th} Tube in Vertical Column of Bundle

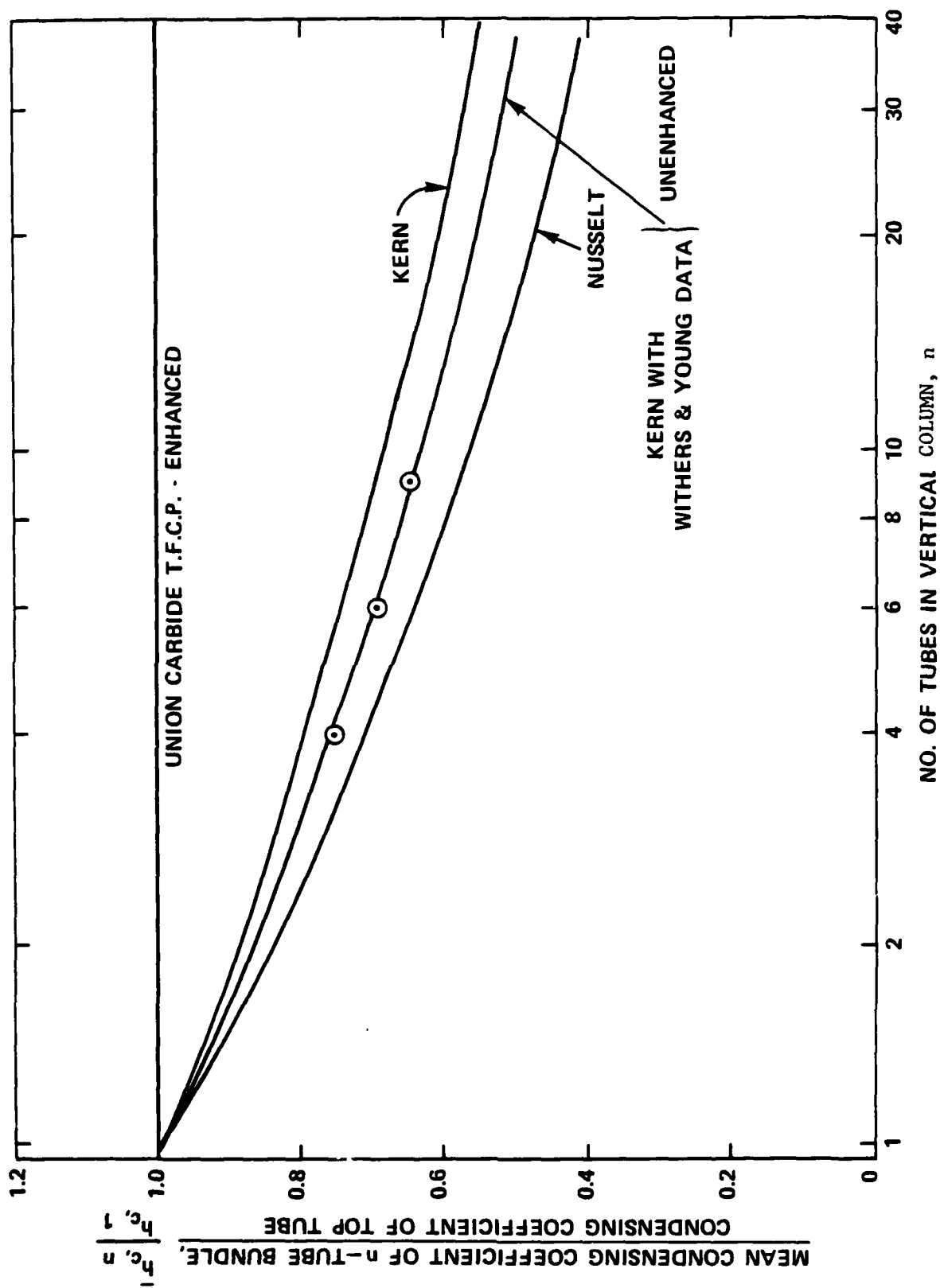


Figure A-4 - Correction Factor for the Mean Steam Condensing Coefficient for a Bundle Containing n Tubes in a Vertical Column

In conclusion, when the T.F.C.P. is used, the bundle condensing performance is not reduced due to condensate loading from tubes above, while the plain-tube bundle condensing heat transfer is only 43 to 57 percent that of the top tubes.

LIQUID LOADING/STRIPPING TESTS

In order to provide a visual demonstration of the surface-tension effect due to the presence of the T.F.C.P., a very simple test rig was constructed by Union Carbide. It basically consisted of two 18-in. (45.7-cm)-long horizontal tubes separated by 2 in. (5.08 cm) center-to-center vertically. The top (overhead) tube (5/8-in. (1.59-cm) dia.) contained many small orifices (1/32 in. (0.794 mm) dia.), equally spaced (0.3 in. (0.762 cm) center-to-center) axially, through which water was discharged uniformly and allowed to "rain" onto the bottom tube. To promote a uniform drip rate axially, a wire was wrapped around the top tube so that each orifice was located between coils. The bottom, or test tube, was cut from the Navy-supplied tubing and had three differently configured external surface sections:

- (1) The first 6-in. (15.2-cm) section was spirally-roped with the T.F.C.P. (titanium wire),

- (2) The second 6-in. (15.2-cm) adjacent region was spirally-roped without the T.F.C.P., and

- (3) the last section was plain.

The perforated drip tube was connected to a hot-water supply which was flow-regulated to simulate condensate loading from 21 vertically stacked tubes onto the test tube under equivalent heat fluxes of 20,000 and 40,000 Btu/hr-ft² (63 and 126 kW/m²).

By careful observations through a large magnifying glass, it was noticed that a single drop, upon hitting the plain-tube section, flattened and "fanned-out" on top of the tube over approximately 1.25 in. axially.

A similar "fanning-out" occurred on the roped section over approximately 1.25 in. (4 pitches), regardless of whether the drop landed on or off a groove.

However, when liquid droplets fell onto the T.F.C.P.-enhanced section, the "wetting" area was much smaller: when the drop hit the wire, only two pitches were involved (i.e., 0.6 in. (1.52 cm), but when the drop fell midway between the wires, 3 pitches (i.e., 0.9 in. (2.29 cm) were affected, presumably caused by wire splash-over on either side of the droplet.

It was also observed that the T.F.C.P. rapidly "stripped" the water film between the wires to a level much below that noted on the other sections, and the wires effectively facilitated "pulling/stripping" the liquid from the top surface, as expected. These stripping effects were even more pronounced as the liquid loading (flowrate) was increased.

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A-2 Withers, T.G. and Young E.H., "Investigation of Steam Condensing on Vertical Rows of Horizontal Corrugated and Plain Tubes," Paper No. 5 of a Symposium on Enhanced Tubes For Desalination Plants, March 11-12, 1969.

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